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NOTES ON MACHINE DESIGN

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NOTES ON MACHINE DESIGN

NOTES ON
MACHINE DESIGN

PREPARED BY THE DEPARTMENT OF MARINE
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UNITED STATES NAVAL ACADEMY
FOR THE INSTRUCTION OF MIDSHIPMEN

COMMANDER F. W. BARTLETT, U. S. NAVY

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NOTES ON MACHINE DESIGN

THE DESIGNING COURSE.

DATA DIFFERING FROM THAT IN THE TEXT WILL BE GIVEN FOR THE PROBLEMS.

Calculations of the problems will be made on loose paper and smooth copies made in ink on the interleaved pages. For purposes of reference, important headings will be placed in the margin at left of page. Working, free-hand pencil sketches, properly dimensioned, will be made on the interleaved pages. From the working sketches made, the required views of the problems will be pencilled on the drawing paper to proper scale; all the dimensions properly placed in pencil; and the drawing completed for tracing.

Methods of use of instruments and of work generally will be given great weight in this course.

The drawing paper will be secured on the boards with thumb tacks and centered, and the cutting, border and working lines drawn. Dimensions of these lines: 18″ × 24″, 16″ × 22″, 15″ × 21″ respectively. Slight variations may be made, as required, according to judgment and so that a larger scale may be used.

There will be one problem on each sheet.

Instructions in regard to placing the legend, the number of views required, etc., will be given with each problem.

After the drawings are completed in pencil, they will be traced, and two blue prints made.

PROBLEM I.

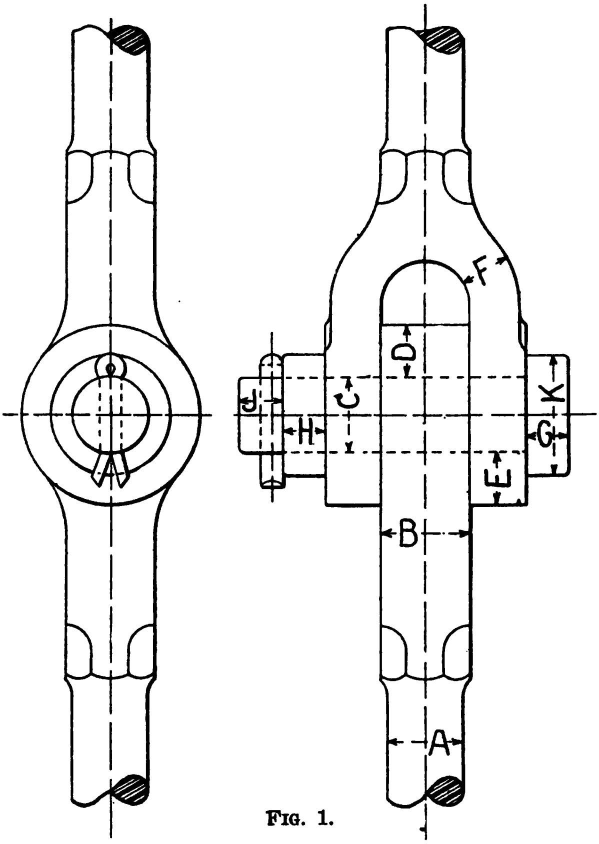

FIG. 1.

Design a knuckle joint for a stay rod of a boiler. Area supported 12″ × 12″. Pressure per gauge, 150 lbs. Material, wrought iron. $f_t = 7{,}000$ lbs. per sq. in.

DRAW FRONT AND SIDE ELEVATIONS: SCALE, FULL SIZE.

To find the diameter of the rod A, which is in tension, we have:—

$P = 12 \times 12 \times 150 = 21{,}600$ lbs. Area of rod $= \frac{21{,}600}{7{,}000} =$ 3.08 sq. ins.

And A, the diameter, $= 1''.98$, or $2''$.

Let B be 1.1A (*Unwin,* p. 210), or,

$B = 1.1A = 1.1 \times 2'' = 2\frac{1}{4}''$, nearly.

To find the diameter of the pin C, which is calculated to resist bending. The pin is in the condition of an uniformly loaded beam $2\frac{1}{4}''$ long, supported at each end. We have, therefore,

$$M = \frac{wl^2}{8} \text{ and } w = \frac{P}{l}. \text{ Hence, } M = \frac{Pl}{8} = \frac{21{,}600 \times 2\frac{1}{4}''}{8} = \frac{21{,}600 \times 9}{8 \times 4}.$$

Also, $M = f_t z$ where $z = \frac{\pi d^3}{32} = .0982\ d^3$ and $f_t = 7{,}200$ lbs. per sq. in.

Haswell gives the bending strength of a bar of wrought iron, 1 foot long and 1″ square as 600 lbs. For a bar of the same cross-section and 1″ long, $f_t = 7{,}200$ lbs.

$$\text{Therefore, } \frac{21{,}600 \times 9}{8 \times 4} = 7{,}200 \times .0982\ d^3, \text{ or}$$

$$d = \sqrt[3]{\frac{21{,}600 \times 9}{8 \times 4 \times 7{,}200 \times .0982}} = 2''.06.$$

Then C, or diameter of the pin, is 2″.

To find D, the thickness of the part surrounding the pin. The metal is in the condition of an uniformly loaded beam fixed at both ends.

Fig. 2 shows a perspective view of the eye of the bottom rod and of the pin through it. It is understood that the eye is pulling down upon the pin. The portion of metal considered as a beam is that which would shear out if the eye should fail in that way. Two planes shown by dotted rectangles form the ends of the beam where it is "fixed" to the remaining part of the eye. The little beam is drawn separately above, the "fixed" planes at the ends being shown by ruling them with vertical lines.

$$l = \text{length} = C = 2''. \quad M = \frac{Pl}{12} = \frac{21{,}600 \times 2}{12} = 3{,}600.$$

Also, $M = f_t z$ where $f_t = 7{,}200$ and $z = \dfrac{bh^2}{6}$. $b = 2\frac{1}{4}''$.

Therefore, $M = \dfrac{7{,}200 \times h^2 \times 9}{6 \times 4} = 2{,}700\ h^2 = 3{,}600$, and $h = 1''.155$, or $D = 1\frac{3}{16}''$.

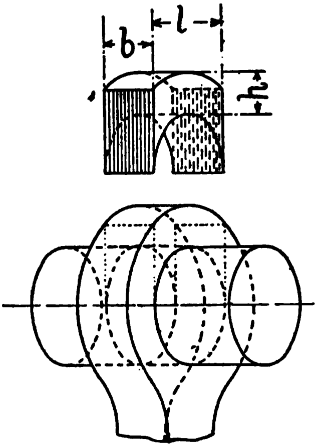

FIG. 2.

The depth E is taken equal to D for symmetry and is thus made amply strong to resist irregular stresses and fitting.

As possibly two-thirds of the stress may come on each arm of the split end, due to irregular fitting, F is made about $\frac{2}{3}$ of B, or $1\frac{1}{4}''$.

As one dimension of the cross-section at F is the same as one side of the square section at B, the other dimension alone takes the entire variation.

The lower portions of the forked ends are made a little wider than at F to make a good bearing for the pin, or $1\frac{1}{2}''$. Or, from *Unwin*,

$$.75 \times \text{diameter of pin} = 1\tfrac{1}{2}''.$$

G, the thickness of the head of the pin, is taken as .5 C (see *Unwin*).

K, the diameter of the head of the pin, is 1.5 C = 3″.

H, the collar, has the same dimensions as the head of the pin.

J is taken long enough for a round split pin of a diameter $= \dfrac{C}{4}$ and an equal amount beyond. This is empirical.

The space above D is taken a little greater than D, or $1\frac{1}{2}''$. The eight-sided portions at top and bottom are empirical and represent the corners of the square head bevelled off to make an eight-sided section. The dimensions of this portion depend upon the

work in the blacksmith shop, as a fuller is used to shape the octagon. Make the length that required for use of proper fuller.

The part where the cylinder merges into the octagonal prism must be carefully drawn to show that it is *octagonal* not *hexagonal.* Draw, in pencil only, a plan like that in Fig. 3, draw the octagon and its inscribed and circumscribed circles. On the front elevation, assume some fairly large radius r for the arc of enlargement, a b, of the cylinder, and draw the arc, extending it to g. From g′ on the plan project down to g on this arc. Pass horizontal lines through b and g. The lines bf and gk represent horizontal planes which intersect the surface of revolution generated by the arc a b g, in circles. These circles are seen in their true form on the plan as the inscribed and circumscribed circles already drawn. Corresponding points, therefore, can be projected from plan to elevation as b′b, h′h, c′c, i′i, etc. From i and j draw i n and j p, the near edges of the octagonal prism. Find by trial arcs of circles to connect h c i, i d j and j e k, as shown.

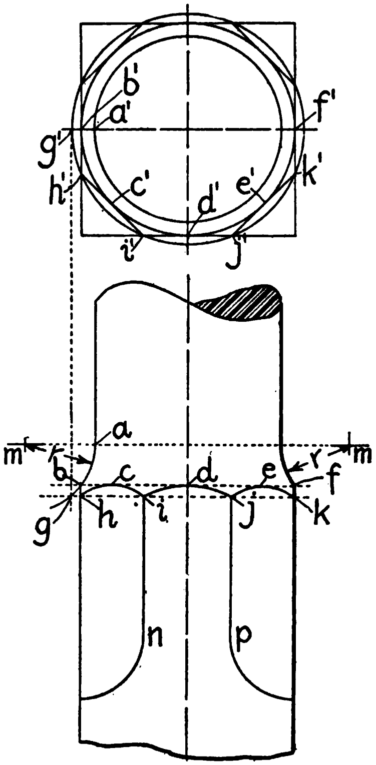

FIG. 3.

The fact that b and f are on a level with c, d, e, and are *not the same points as h and k,* shows the difference between octagon and hexagon. From n and p any arcs may be drawn to the side boundaries.

PROBLEM II.

Design a screw jack to raise a load of fifteen tons. Height of lift 12″. The frame is to be of cast iron; the screw, head, nut and band of wrought iron. The head of the screw is to have two $1\frac{1}{4}''$ holes at right angles to each other for the turning bar. There is to be a cast-iron swivel plate on top of the head. The screw must not overhaul.

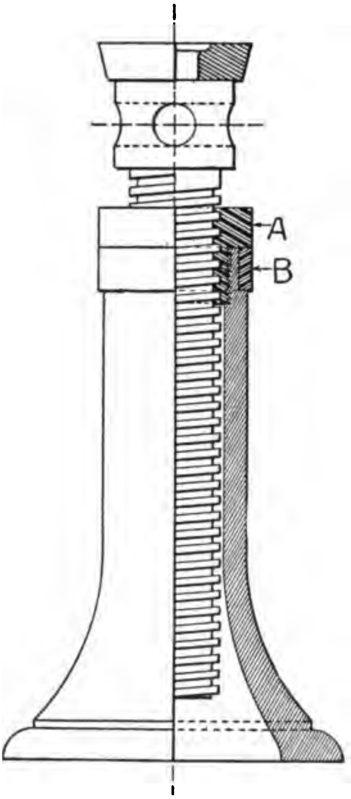

FIG. 1.

1. *To find the diameter of the screw.*—This is calculated for the bottom of the thread, (as the threads do not assist in resisting the compression due to the load).

Let A = area of cross-section of screw;

W = total load $= 15 \times 2{,}240$ $= 33{,}600$ lbs.;

M = Pl = turning moment required to raise W.

This moment will induce a torsional stress $f_s = \frac{yM}{I} = \frac{M}{Z}$, which must be combined with the pure compressive stress $f_c = \frac{W}{A}$, and this combined stress for wrought iron must not exceed 10,000 lbs. per square inch, (assuming that the screw is frequently loaded and unloaded, and not subject to shock or reversal of stress).

In the preliminary calculations, we will assume that an increase of 20% in the area (over that required for pure compression alone) will be a sufficient allowance for the effect of the torsional stress f_s, or

$A = \frac{33{,}600}{10{,}000} \times \frac{1.2}{1} = 4''.032$ and $d_2 = 2''.28$ or $2\frac{1}{4}''$, and A becomes = 4 sq. ins.

(Fig. 1 shows a half sectional view of the screw jack with all parts assembled.)

2. *To find the dimensions of the threads.*—For a screw which will not overhaul, the pitch angle must be less than the angle of friction, *i. e.* $\tan \alpha < \tan \phi$. (See note 1.)

Threads will be made square for the best transmission of power and for durability. To be safe against overhauling with the materials used and good lubrication, μ must not be given a greater value than 0.10.

$\therefore \phi = \tan^{-1} \frac{0.10}{1} = \tan^{-1} \frac{1}{10} = 5^\circ\ 45'$, and $\alpha < \phi$.

From $\tan \alpha = \frac{p}{2\pi r}$, substituting proper values (see note 2), and solving we get $p = 0.84$, which is the maximum value to satisfy the conditions that the screw will not overhaul.

It would be desirable to modify the value of p so that the thread could be conveniently cut in a lathe.. It is obvious that p cannot be increased without increasing the diameter of the screw, else α will have a greater value than is allowable.

From *Unwin,* pitch $= p = .16d + .08 = .48''$, and as it will increase the mechanical power and be more economical to reduce p, the nearest even value to $.48''$ will be selected, or $= \frac{1}{2}''$.

3. *To check the calculations for cross-section of screw.*—The moment required to raise the load W is equal to

$$M = Pl = Hr = Wr\left(\frac{p + 2\pi rc}{2\pi r - pc}\right),$$ (see note 1) which equals

$$33{,}600 \times 1\tfrac{1}{4}\left(\frac{\frac{1}{2} + 2 \times \frac{22}{7} \times 1\frac{1}{4} \times \frac{1}{10}}{2 \times \frac{22}{7} \times 1\frac{1}{4} - \frac{1}{2} \times \frac{1}{10}}\right) = 6{,}994 \text{ in. lbs.}$$

This is the moment which produces the torsional stress $f_s = \frac{M}{Z}$ $(Z = \frac{\pi d_2^3}{16})$, which equals

$\frac{6{,}994 + 7 \times 16}{22} \times \left(\frac{4}{9}\right)^3 = 3{,}128$ lbs. per sq. in., and this torsional stress must be combined with the pure compressive stress

$$f_c = \frac{W}{A} = \frac{33,600}{4} = 8,400 \text{ lbs. per sq. in.,}$$

in order to obtain the combined stress f on the screw which is given by the formula

$$f = .35\ f_c + .65\ \sqrt{f_c^2 + 4f_s^2} = .35 \times 8,400$$
$$+ .65\ \sqrt{(8,400)^2 + 4\ (3,128)^2} = 9,749 \text{ lbs.}$$

As this result is well within the limit of safety the area of cross-section of screw previously calculated will be taken.

4. *To find the dimensions of the nut.*—The height of the nut is determined from the equation

$$W = Kn \times \frac{\pi}{4}\ (d_1^2 - d_2^2),$$

in which K is the maximum allowable pressure in lbs. per sq. in. of bearing surface, and its value depends upon the speed.

For a rubbing velocity of less than 50 feet per minute (and it is fair to assume that the screw will never exceed this limit), the value for wrought iron is K = 2,500, ∴ the number of threads required in the nut

$$n = \frac{W}{K \times \frac{\pi}{4}\ (d_1^2 - d_2^2)} = \frac{33600}{2500} \times \frac{28}{22} \times \frac{2}{5} = 6.85 \text{ nearly.}$$

Height of nut equals p × n = ½ × 6.85 = 3.425 or 3½″.

(A short calculation will show that the direct shearing stress on the threads will come well within the limit allowed for wrought iron, with ample allowance for wear.)

The nut is screwed into the inside of the frame with a pipe thread which need not be calculated as the nut is screwed hard up against the shoulders a and b and against the band at c, Fig. 3. B is a wrought-iron band shrunk on the frame to prevent the nut and frame from spreading. The dimensions are empirical, and must be decided upon for each size of jack. For this size of jack the band is ¼″ thick and 1½″ long. The outside diameter of band = the outside diameter of the nut = the outside diameter of frame + ¼″.

5. *To find the dimensions of the head of the screw.*—The area remaining after the holes are taken out is made equal to the area

of the screw at bottom of thread as it must sustain about the same stress.

(Fig. 2 is a horizontal section taken at the middle of the holes.) Let x = the diameter required,

$$A = \frac{\pi}{4} x^2 = 4\ (1\tfrac{1}{4} \times 2x - 1\tfrac{1}{4} \times 1\tfrac{1}{4})$$

(since two lengths 1¼″ across, less the central portion 1¼ × 1¼, are removed).

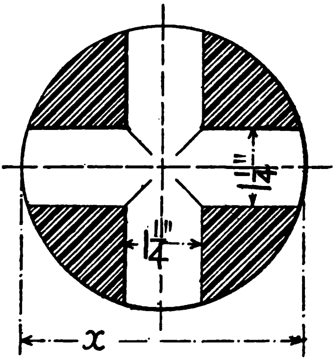

Fig. 2.

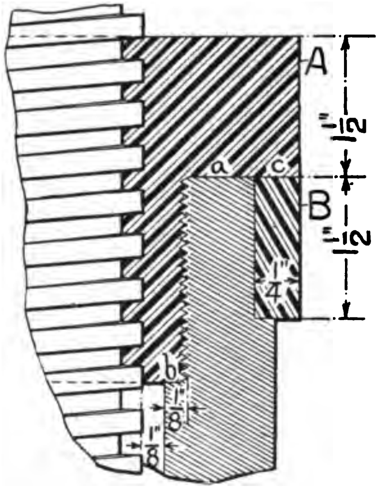

Fig. 3.

Solving, we find

$$x = 3.86 = 4'', \text{ nearly.}$$

The depth of the head below the holes is calculated on the assumption that the end of the screw tends to punch upward in to the head, and that the resistance to this is one of pure shear (which is on the safe side, as there is an additional resistance to crushing offered by the small centrally projecting portions in the vicinity of the holes).

Let t, Fig. 4, = thickness required such that

$$\pi d_2 t f_s = W,$$

where f_s is the maximum shearing stress for wrought iron = 7,500.

Solving, we find

$$t = \frac{W}{\pi d_2 f_s} = \frac{33{,}600}{\frac{22}{7} \times 2\frac{1}{4} \times 7{,}500} = 0.633 = \frac{5}{8}'', \text{ nearly.}$$

It is customary to make the distance from the top of the hole to the top of the head the same as that below $= \frac{5}{8}''$,

$\therefore$ total depth of head $= 2 \times \frac{5}{8} + 1\frac{1}{4} = 2\frac{1}{2}''$.

6. *Details of the swivel plate.*—This is arranged to turn by dropping it into place and then slightly upsetting the central pin over the top of the plate. The central pin is a part of the screw head, and for this size of jack is taken as $1\frac{1}{2}''$ diameter. It stops just below the level of the top of the swivel plate, the latter being countersunk. Height of swivel plate $= 1''$; upper diameter $= 5''$; and lower diameter $= 4\frac{1}{2}''$. Depth of counter bore $\frac{3}{16}''$; diameter of counter bore $2''$. These dimensions vary according to the size of jack. By calculation the bearing surface may be worked out and compared with the crushing strength of cast iron. It is ample in this case.

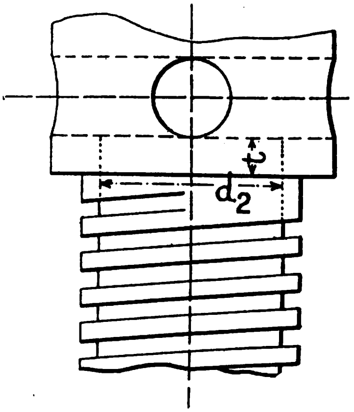

FIG. 4.

7. *Details of the frame.*—The diameter of the inside of the frame is that of the outside of the screw plus $\frac{1}{8}''$ to $\frac{1}{16}''$ clearance on each side. In this case it is $2\frac{3}{4}'' + \frac{1}{4}'' = 3''$. (Area corresponding $= 7.08$ sq. ins.) The outside diameter is found from:

Area of inside opening + area of screw at bottom of thread = area of outside diameter. "Considering the crushing strength of the cast iron of the frame equal to that of the screw."

Or $7''.08 + 4'' = 11.08$ sq. ins., and

Outside diameter $= 3''.74 = 3\frac{3}{4}''$.

Thickness of frame $= \frac{1}{2}$ (outside diameter — inside diameter) $= \frac{3}{8}''$.

This is for strength only, and if used it would be necessary to increase the thickness at the top of the frame for the threaded portion of the nut; also, in order to avoid the danger of breaking accidentally, and to insure a better casting the thickness is increased to ⅝″.

The outer diameter of frame is then = 3″ + 1¼″ = 4¼″.

The diameter of the base of the frame is made about 2½ times the above or 10½ for a good spread to prevent tipping, and for a larger surface. The base is made thicker in section so as to press on a greater quantity of material.. It is finished in some such manner as shown in sketch. The bottom of the frame must be at least ½″ below the bottom of the screw when at its lowest position in order to protect the screw.

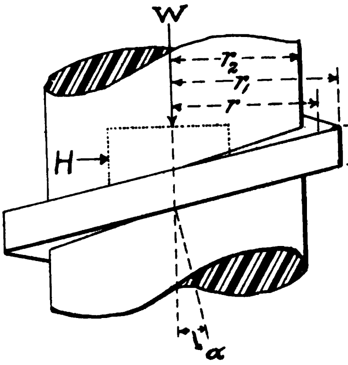

Fig. 5.

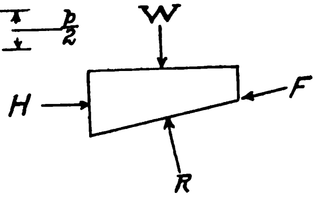

Fig. 6.

The inside diameter of the frame at the upper end is counterbored ⅛″ all round for a length of 2″ and threaded for the nut, terminating in a square shoulder against which the bottom of the nut is screwed up hard. Also the outside diameter at the top of the frame is turned down ⅛″ all round (over a length of 1½″) to a square shoulder on which the band rests.

Note 1.—*Analysis of the action of screw.*—Referring to Fig. 5, the turning of the screw relative to the nut may be considered equivalent to moving a load W on an inclined plane whose angle with the horizontal is the same as the mean pitch angle a of the thread.

Let r_1 = outside radius of thread.
r_2 = inside " " "
r = mean " " " $= \frac{r_1 + r_2}{2}$, approximately.
p = pitch of thread.
a = mean pitch angle, *i. e.* $\tan a = \frac{p}{2\pi r}$.
μ = coefficient of friction between nut and thread.
ϕ = angle of friction, *i. e.* $\tan \phi = \mu$.

Now, in order to raise W, consider that it is a free body moving uniformly up an incline, under the system of forces shown in Fig. 6, where W = axial load; R = the normal reaction between nut and thread due to W; H = the horizontal push forcing the load up the incline and F the friction in direction of the incline due to normal pressure R, then it is evident that

$$R = \frac{W}{\cos a},$$

$$F = \frac{\mu W}{\cos a},$$

$$H = W \tan (a + \phi),$$

and the turning moment

$$M = Hr = Wr \tan (a + \phi).$$

Since $\tan a = \frac{p}{2\pi r}$, and $\tan \phi = \mu$, while

$$\tan (a + \phi) = \frac{\tan a + \tan \phi}{1 - \tan a \tan \phi}.$$

$$M = Hr = Wr\left(\frac{p + 2\pi r\mu}{2\pi r - p\mu}\right).$$

NOTE 2.—From, $\tan a = \frac{p}{2\pi r}$, we have $p = 2\pi r \tan a$.

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Remembering that for square threads, depth of thread $\frac{19}{40}p = \frac{p}{2}$ nearly, and that r_2 = radius at bottom of thread, and $r = r_2 + \frac{p}{4}$,

$$p = 2\pi\left(r_2 + \frac{p}{4}\right),$$

and substituting

$$p = \frac{2\pi r_2 \tan\alpha}{1 - \frac{\pi}{2}\tan\alpha}.$$

NOTE 3.—Regarding the efficiency of the square screw thread to transmit power:

$$\text{Efficiency} = E = \frac{\text{Useful work}}{\text{Total work}} = \frac{Wp}{2\pi r H}.$$

$$\text{For 1 turn} = \frac{W \times \frac{p}{2\pi r}}{H} = \frac{W\tan\alpha}{W\tan(\alpha+\phi)} = \frac{\tan\alpha}{\tan(\alpha+\phi)}.$$

(See note 1, Figs. 5 and 6.)

From this it appears that E becomes 0 for $\alpha = 0$, and for $\alpha = 90° - \phi$, and must therefore have a maximum value between these limits. To determine this maximum, write

$$\frac{\tan\alpha}{\tan(\alpha+\phi)} = \tan\alpha\cot(\alpha+\phi).$$

Take first differential and equate = 0,

$$\therefore \frac{\cot(\alpha+\phi)}{\cos^2\alpha} - \frac{\tan\alpha}{\sin^2(\alpha+\phi)} = 0.$$

Solving, we find $\alpha = 45° - \frac{\phi}{2}$, and substituting above

$$\text{Maximum } E = \frac{\tan\left(45° - \frac{\phi}{2}\right)}{\tan\left(45° - \frac{\phi}{2} + \phi\right)} = \frac{\tan\left(45 - \frac{\phi}{2}\right)}{\tan\left(45° + \frac{\phi}{2}\right)}.$$

Regarding the efficiency in this case,

If $\alpha < \phi$ the load will not overhaul,

$\alpha = \phi$ we have a condition of equilibrium,

$\alpha > \phi$ the load will overhaul.

Now, for a screw which does not overhaul, it becomes evident that the limiting value of α is ϕ, and the maximum efficiency, substituting is

$$\frac{\tan \phi}{\tan 2\phi} = \frac{1 - \tan^2 2\phi}{2} = 0.5 - 0.5 \tan^2 2\phi.$$

This can never exceed 0.5 or 50%.

For V threads,

$$M = Pl = Hr = Wr\left(\frac{p + 2\pi r\mu \sec \beta}{2\pi r - p\mu \sec \beta}\right)$$

(see Fig. 7).

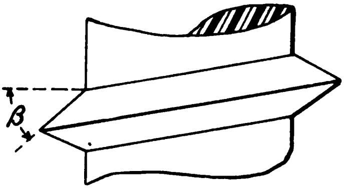


FIG. 7.

$$\text{Efficiency} = \frac{\tan \alpha\,(1 - \mu \tan \alpha \sec \beta)}{\tan \alpha + \mu \sec \beta}.$$

Comparing these with similar equations for square threads, it is at once evident that considering efficiency and mechanical power, square threads should be used in preference to V threads.

ON THE THEORY OF THE CONNECTING ROD.

The connecting rod of a steam engine is subjected to rapidly alternating, and nearly equal, tensions and compressions. Theoretically it may fail in one of three ways:

1st. As a "long column" in compression, by bending out the line of its axis, *i. e.,* "buckling."

2d. As a "tie rod" by pulling apart under tension.

3d. As a "short column" in compression, by the metal breaking down or "crushing."

Practically we need consider the rod only as a long column in compression, because the formula for strength against failure by buckling gives a larger section than for either of the other two cases.

GORDON'S FORMULA.

The formula for long columns, known as Gordon's formula, has been given in a great variety of forms. One of the most general forms for a column "free at both ends" has been used in the Bureau of Steam Engineering for the *tabulation* of the designs of the connecting rods of all the U. S. naval vessels.

It can also be used for *designing* rods, or special formulæ deduced from it for special cases.

It is:—

$$W = \frac{\frac{C}{N} \times A}{1 + \frac{C}{\pi^2 E} \times \frac{l^2}{k^2}}.$$

An intelligent use of it requires an understanding of the terms and the values given to them under different circumstances.

A, the area, and k, the radius of gyration, of the cross-section of the rod are the unknown factors whose values are to be found by the application of the formula for one or more places along the length of the rod.

W, the load the rod is designed to bear, is sometimes determined by a method given in *Seaton,* p. 173. The Bureau now uses a simpler method, as follows: It is supposed that on starting the engine, before the receiver spaces have been filled with steam, the piston of the H. P. cylinder may be subjected for one or two strokes to full boiler pressure on the upper surface, while the lower surface is under atmospheric pressure only. R, the thrust on the H. P. piston rod is then the area of the H. P. piston × boiler pressure. W, the thrust on the connecting rod is greater than R by a small percentage. The difference is a maximum when the crank is at right angles to the axis of the cylinder and the connecting rod in consequence makes its greatest angle with that axis. This angle is of course $\sin^{-1}\frac{\text{crank length}}{\text{connecting rod length}}$ (see Fig. 1) and with the proportions there given is $\sin^{-1}\frac{24''}{96} = \sin^{-1}\frac{1}{4} = 14^\circ\ 29'$. At the moment considered, the cross-head pin is acted on by three forces, W, the thrust of the connecting rod; R, the thrust of the piston rod; and S, the resistance of the guides (see Fig. 2). Neglecting friction, *i. e.*, taking S perpendicular to R and resolving forces horizontally, we have $W \cos\theta = R$, or $W = R \sec\theta$.

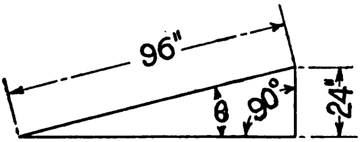


Fig. 1.

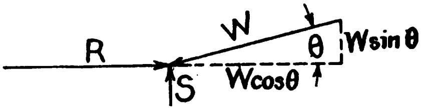


Fig. 2.

In marine practice the crank is very frequently one-fourth the length of the connecting rod. In all such cases

$$W = R \sec 14^\circ\ 29' = R \times 1.033.$$

This calculation applies only to the H. P. cylinder, but in nearly all cases the rods are made equal and interchangeable.

The factor $\frac{C}{N}$ is often written as one symbol, f, and is defined as the "safe working fiber stress." However, since C also appears separately, the older method of using C, the "ultimate strength" of the material, and providing a "factor of safety" N is adhered to as the best in this case.

The ultimate strength of the metal used is largely a matter of cost, and constantly changes with improved methods of manufacture. For slow-running or small-sized engines, where saving in weight is not of prime importance, wrought iron, on account of its cheapness is still used. For quick-running engines the tendency has been towards the best attainable material. This tendency is

well shown by the specifications of our naval vessels here briefly tabulated.

Name of material.	Ult. str.	Vessels.
Wrought iron	36,000	Puritan and Miantonomoh.
Mild steel	55,000	Boston to Dolphin.
Steel	65,000	Maine (old) to Iowa.
Oil-tempered steel	80,000	Kearsarge to Illinois and destroyers.
Oil-tempered nickel steel..	95,000	Maine (new) and all later vessels.

The first steel used was of very low carbon, 0.20%, for the sake of great ductility and consequent safety against fracture from sudden shock.

The process of oil-tempering has enabled steel of 0.30 to 0.40% carbon to be used, with its attendant great strength and without loss in ductility. It is requisite, however, that the thickness of the metal tempered should nowhere exceed 2½″ to 3″, and consequently *for large rods oil tempering necessitates a hollow section.*

The presence of a small percentage of nickel in the steel, makes it possible to reach an ultimate strength of 95,000 lbs. Without it, it is barely possible to get 80,000 lbs.

The factor of safety must be a large one by reason of the "vibrating load" on the rod (*Unwin,* p. 34). Its value usually lies between 8 and 10. Recently the Bureau of Steam Engineering has used values of 8.5 and 8.9.

π is of course 3.1416. π^2, in Gordon's formula, is however taken as 10 for simplicity.

E is the modulus of elasticity of the metal of the rod. For all iron and steel rods its value is taken as 30,000,000 lbs.

l, the length of the column when the formula is used to determine the section at the middle of the rod is the length "from center to center."

THE ROD CONSIDERED AS A COLUMN.

The strength of a column to resist buckling is known to depend largely on the nature of the "end fastenings." The connection of the rod to the cross-head at one end and to the crank at the other is of such a kind as to allow perfect freedom of adjustment in the "*plane of motion of the rod,*" while very little, if any, is permitted in that *at right angles to the plane of motion.*

In theory, therefore, the bending of the rod in the two planes should be treated as two different cases. The formula we have assumed applies directly to buckling in the plane of motion of the rod. It can be applied also to determine the strength against buckling in the other plane by substituting $\frac{l}{2}$ for l, on the ground that the strength of a column "fixed at both ends" is equal to that of one of half its length "free at both ends." One other change must be made. The radius of gyration employed is always that for the section of the rod about an axis in the plane of the section *perpendicular to the plane in which bending takes place.*

Since all other factors are unaltered, if the value of W in the second application of the formula is not to change, the alteration in k must counterbalance that in l. If k changes to $\frac{k}{2}$ this condition is evidently fulfilled. Therefore, *to give equal strength against both kinds of buckling the radius of gyration of the section about an axis in the plane of motion must equal half the radius of gyration about an axis perpendicular to the plane of motion.*

For any circular, or flattened circular section, solid or hollow, likely to be used for a connecting rod, the two radii of gyration will be so nearly equal that we shall have a great excess of strength against bending sidewise.

For a rectangular section the two radii are proportional to the sides to which they are parallel, in other words to the sides perpendicular to the axes considered. If it is taken as a rule that the thickness of the section at right angles to the plane of motion shall equal or exceed half that in the plane of motion, bending sidewise will again be provided against.

Thus no double application of the formula is ordinarily required and the only k considered is that for bending in the plane of motion of the rod.

TAPERING THE ROD.

The rod does not need to be of equal section throughout its length. Were it really a column pure and simple it could be made to taper in both directions from the middle. The section at any point along the rod at a distance x from one end can be obtained as follows:—The piece of rod of length x forms a column "fixed at one end" (*i. e.* fixed or made fast by the molecular forces which bind it to the other part of the rod) and "free at the other." Such a column is equivalent to one of twice its length "free at both ends." The section at a distance x from one end is found therefore by putting 2x in place of l in our formula.

If the sections at a great number of points are determined the resulting column will be one swelling gradually at the middle like an ornamental architectural column. Rods were at first so designed. More frequently the sections at two points near the ends were determined and the rod made to taper uniformly from the middle in both directions. Many slow-moving rods are still so made.

With the increase of speed of engines it has been found desirable to increase the strength of the rods near the crank end. This is due to the additional bending stresses set up by the "transverse inertia" of the connecting rod itself. These stresses increase rapidly with the speed of the engine and are a maximum in the part of the rod near the crank end where the cross motion is greatest.

The tapering of the rod at the crank end was first given up, leaving that portion of the rod of parallel section. Finally the practice has come in of carrying on the taper, or rate of increase, which is determined by the sections at the cross-head neck and the middle of the rod, to the crank end. This practice will be followed in the succeeding pages.

THE ROD CONSIDERED AS A TIE ROD.

The formula for the strength of a tie rod or short column is $W' = \frac{C}{N} \times A$. This expression is the numerator in the formula for the strength of the rod as a column and we note also that the denominator is *necessarily greater than unity.* W′ will therefore always exceed W.

PROBLEM III.

TO DESIGN A CONNECTING ROD WITH STRAP END, FOR A SLOW-MOVING ENGINE.

Specifications and Data.—Compound Engine. Diameter of H. P. cylinder 30″. Stroke 36″. Boiler pressure 80 lbs. Material of connecting rod, wrought iron. Length between centers, 90″. Dimensions of crank pin, 8½″ × 10″. Assumed ratio of length to diameter of connecting rod, 15. Assume a solid circular section for the rod.

Formula.—Since k^2 for a solid circular section equals $\frac{d^2}{16}$, the formula can be written

$$W = \frac{f \times \frac{\pi}{4} d^2}{1 + \frac{16C}{\pi^2 E} \times \frac{l^2}{d^2}}, \text{ or } d = \frac{\sqrt{W\left(1 + \frac{16C}{\pi^2 E} \times \frac{l^2}{d^2}\right)}}{\sqrt{\frac{\pi}{4} f}}.$$

The factor $\frac{l^2}{d^2}$, is often written r^2, where r is the ratio of length to diameter given above in the specifications. The factor $\frac{16C}{\pi^2 E}$ is also written as 4a, and $\sqrt{\frac{\pi}{4}} f$, as b. Thus $d = \frac{\sqrt{W \cdot (1 + 4ar^2)}}{b}$.

For ordinary wrought-iron rods, $a = \frac{1}{3,000}$.

When the thrust on the connecting rod, W, is taken as the "working" thrust, as in *Seaton's* method, p. 173, a low value of f is taken to allow for the maximum thrust being greater than the calculated thrust. Then $f = 3,000$ and $b = 48.5$.

When the Bureau's method of using the maximum thrust is followed, f can be taken as 4,000 lbs., and b, therefore, 56.0.

The formula is solved by assuming a value for r, from a knowledge of similar designs, and finding d. Then find the true value of r, as $r = \frac{l}{d}$. This result must agree reasonably well with the ratio r, assumed. If not found to agree, a new trial value of r is taken and a second solution made.

Application of formula to find the middle diameter of the connecting rod.

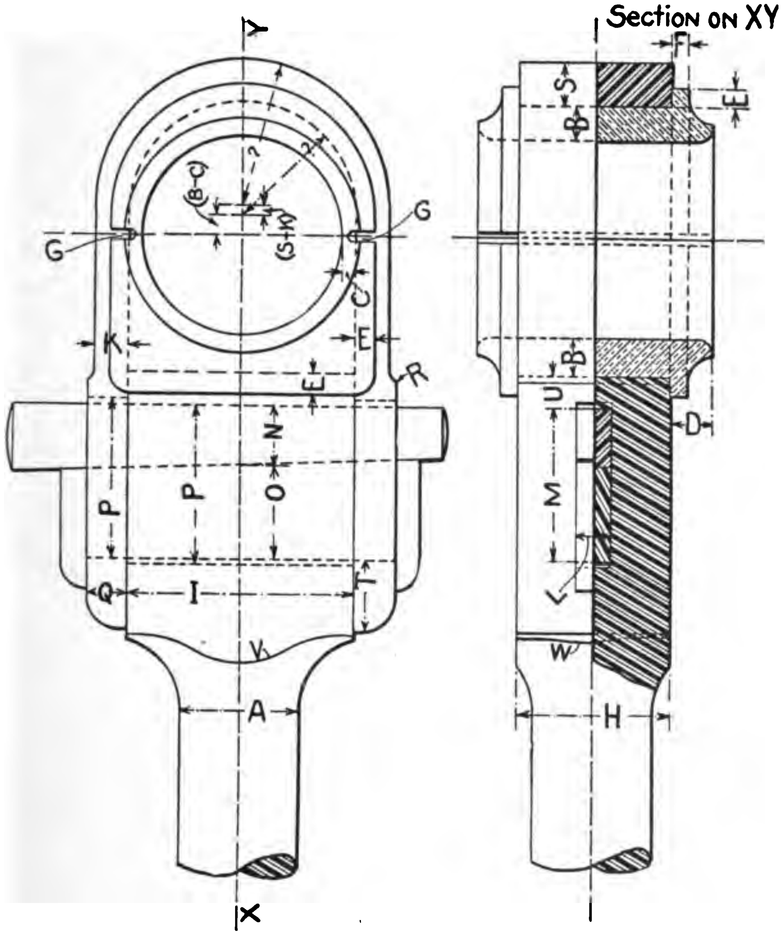


Fig. 3.

R, the thrust on the piston rod is, by the Bureau's method, 706.9 × 80 = 56,552 lbs. $\theta = \sin^{-1} \frac{1}{5} = 11^\circ\ 31'$. $\sec \theta = 1.021$. $W = R \sec \theta = 56{,}552 \times 1.021 = 57{,}700$ lbs.

$$d = \frac{\sqrt{57{,}700 \times \left(1 + \frac{4 \times 15^2}{3000}\right)}}{56} = \frac{\sqrt{57{,}700 \times 1.3}}{56} = 4''.89.$$

This would give a value for the ratios, r, a little over 18, instead of 15 as taken. A second application of the formula, with r = 18,

gives, $d = \frac{\sqrt{57{,}700 \times 1.432}}{56} = 5''.01$. Call this value 5″.

It is sufficiently accurate for a long rod to take the diameter at the ends as .9 × d or 4½″.

Brasses (see *Unwin*, Part II, pp. 108 and 111).—The dimensions of the crank pin, obtained from other calculations, are, 8½″ diameter and 10″ long. The unit for calculating the brasses is found from, $t = .08d + \frac{1}{8}''$, where d is the diameter of the pin.

Then:—unit $= t = .08 \times 8''.5 + \frac{1}{8}'' = ''.80$.

B, the thickness at the bottom, $= 2t = 1''.6 = 1\frac{5}{8}''$.

C, the thickness at the edges, $= .7t = ''.56 = \frac{9}{16}''$.

D, the overhang, = t to 2t.

In this case, in order to reduce the weight of the stub end, by making it narrower, the overhang is taken at $2\frac{1}{2}t = 2.5 \times ''.8 = 2''$.

E, the width of the flange to hold the brass in the strap, $= t = \frac{13}{16}''$.

F, the thickness of the same, $= t = \frac{13}{16}''$.

At G, the brasses are planed away at the edges, as shown, in order to reduce the labor of fitting the brasses.

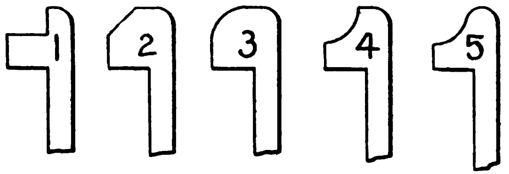


Fig. 4.

The contour of the line joining the end of the overhang and the end of the flange is empirical. No. 5 is generally used in naval designs, as it is amply strong and is lighter than all except No. 1, which is seldom used. In this case the curves at outside and inside are about as drawn in No. 4.

Stub End and Strap.

NOTE.—In calculating the dimensions of the stub end, strap, etc., f_t is taken as 4,000 and f_s as 3,000 lbs. per sq. in.

H, the width of the strap = the width of the stub end = length of crank pin — 2 × overhang = $10'' - 4'' = 6''$.

I, the breadth of the stub end = diameter of pin + 2 × thickness of the brasses at the edges = $8\frac{1}{2}'' + 2 \times \frac{9}{16}'' = 9\frac{5}{8}''$.

The thickness of the strap is found by calculating the dimensions to resist a tensile stress. Then,

K, the thickness of the strap × breadth × $f_t = \frac{1}{2}W$,
or, $K \times 6 \times 4{,}000 = \frac{1}{2} \times 57{,}700$. Whence $K = 1''.20$, or, $1\frac{1}{4}''$, nearly.

Key and Gib.—These are taken at about one-fourth the width of the strap. See *Unwin,* Part I, pp. 175 and 179.

L, the width of the key and gib, $= \frac{6}{4}'' = 1\frac{1}{2}''$.

Then, since (*Unwin,* Part I, p. 176), the gib and key need be calculated only for shear:—

M (breadth of key and gib) × L (thickness) × $f_s = \frac{1}{2}W$,
or, $M \times 1\frac{1}{2}'' \times 3{,}000 = \frac{1}{2} \times 57{,}700$, and $M = 6''.41$, or $6\frac{1}{2}''$.

From *Unwin,* Part II, p. 179,

N = breadth of key $= \frac{3}{8}\,M = \frac{3}{8} \times 6''.5 = 2''.44$, or, $2\frac{7}{16}''$.

O = breadth of gib $= \frac{5}{8}\,M$, or $6\frac{1}{2}'' - 2\frac{7}{16}'' = 4\frac{1}{16}''$.

The taper of the key is $\frac{1}{2}''$ to the foot.

The key is made long enough so that the small end projects from $\frac{1}{2}''$ to $1''$ and the large end from $1''$ to $3''$ past the ends of the gib, depending on the size of the connecting rod.

The thickness and length of the hooked ends of the gib are (*Unwin,* Part II, p. 111) $.8\,\delta$, or, $1\frac{1}{4}''$, where δ is the unit = .15 × diameter of crank pin + .2.

It is necessary for this size of connecting rod to allow $\frac{1}{4}''$ in the lengths of the slots in the stub end and strap for drawing down the key in adjusting brasses. So P, the length of the slots in this case is $6\frac{1}{2}'' + \frac{1}{4}'' = 6\frac{3}{4}''$. The dimensions of the slot are, then, $6\frac{3}{4}'' \times 1\frac{1}{2}''$.

Strap.—Since one-fourth of the strap has been cut out for the key way, it is necessary to thicken it at this point in order to keep the necessary strength. Hence:—

Q, the thickness of the ends of the strap, $= \frac{4}{3} \times 1\frac{1}{4}'' = 1\frac{11}{16}''$.

This enlargement of the thickness begins a short distance above the key at R, and is carried to the ends.

Crown of Strap.—To allow for extra bending stresses, the thickness of the crown, at S, is made 1.2 times the thickness of the strap, or,

$$S = 1.2 \times 1\tfrac{1}{4}'' = 1\tfrac{1}{2}''$$ (*Unwin*, Part II, p. 110).

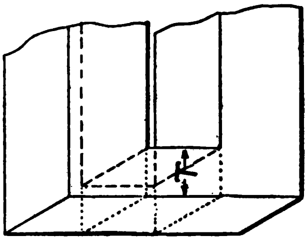


FIG. 5.

The distance T of the key way from the end of the strap is calculated for shearing, the metal being in double shear. Hence:—

$$3{,}000 \times T \times 2 \times 1\tfrac{11}{16}'' = \tfrac{1}{2} \times 57{,}700, \text{ and } T = 2''.88, \text{ or, } 2\tfrac{15}{16}''.$$

The distance U of the key way from the end of the stub end is calculated in the same way:—

$$3{,}000 \times 2U \times 9\tfrac{5}{8}'' = 43{,}725, \text{ and } U = ''.74, \text{ or, } \tfrac{3}{4}''.$$

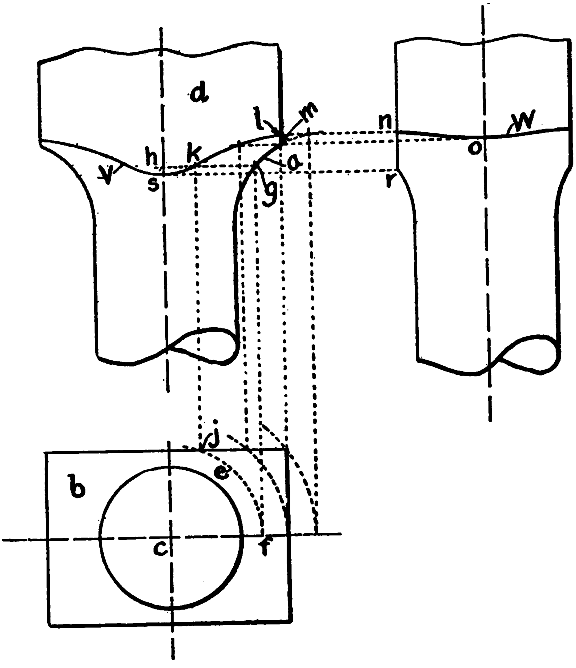


FIG. 6.

To find the curves of intersection of the rectangular stub end with the surface of revolution formed by sweeping the assumed curve of the enlargement of the rod around the central axis:—The curve at *a* is assumed. Make a plan view *b* of the stub end. Pass cylinders with centers at *c* longitudinally through the rod. These will cut the surface swept by *a* in circles concentric with the axis, the projections of which on *d* will be straight lines. They also cut the sides of the rectangle. The intersections of these projections give the points of the curves V and W. The cylinder *e* cuts the curved figure at a circle whose radius is *cf*. Project *f* to *g* and draw *gh*, the projection of the circle cut by the cylinder. *e* also cuts the rectangle at the point *j*.

Project *j* to *k* and one point is determined. *l* and *m* are projected to *n* and *o* for the top and bottom of the curve, W, in the side projection. The remaining points of W are found by using the other diameter of the plan view in the same manner.

PROBLEM IV.

TO DESIGN A CONNECTING ROD FOR A NAVAL VESSEL.

SPECIFICATIONS.

A method of designing a connecting rod of a type much used at present for naval vessels is illustrated in detail in the following pages by giving the calculations for a rod for the battleship Alabama, as an example.

Before beginning the design, certain main dimensions and specifications are decided upon from general considerations of the type of engine and from the design of the adjacent parts.

For the Alabama they are as follows:

Name of ship, : Alabama.
Number of screws, : 2
Type of engines, : Vertical triple expansion.
I. H. P., : 10,000.
Diameter of cylinders, : 33½″, 51″, 78″.
Stroke, : 48″.
Revolutions, : 120 per minute.
Boiler pressure, : 180 lbs.

Material of connecting rod, : Forged nickel-steel, oil-tempered.
Ultimate strength of material, : 80,000 lbs. per square inch.
Factor of safety for rod, : 8.5.
Length between centers, : 96″.
Dimensions of crank pin, : 14¾″ diam. × 17″.
Dimensions of cross-head pin, : 9¾″ diam. × 14″. 5″ bore hole.
Depth of fork from center of cross-head pin, : 19″.
Cross-head pin faced to, : 8⅞″.

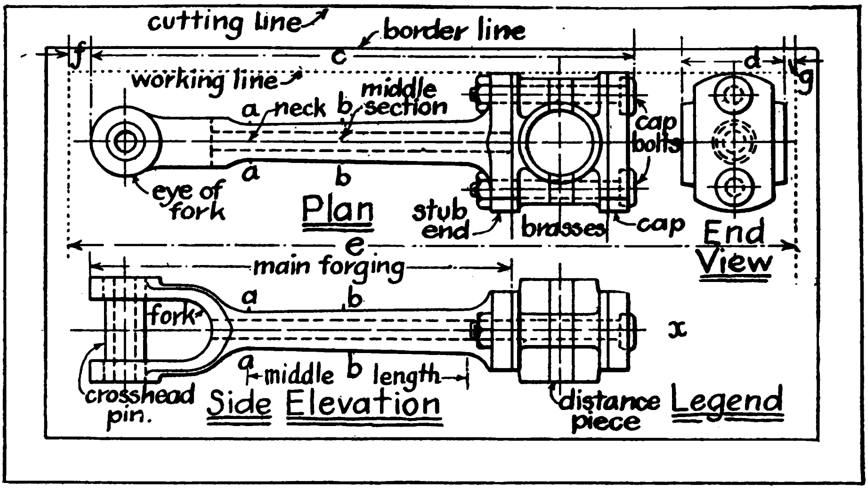


FIG. 1.

THE LOAD THE ROD MUST BEAR.

A general idea of the type of rod to be designed and the names of the parts can be gained from Fig. 1. The sections to be first determined are those at aa and bb. For them Gordon's formula is used and as a preliminary the value of W must be determined from the specifications. Thus:—

R, the thrust on piston rod = area × pressure = 8,814 × 180 = 158,612 lbs., or 158,600.

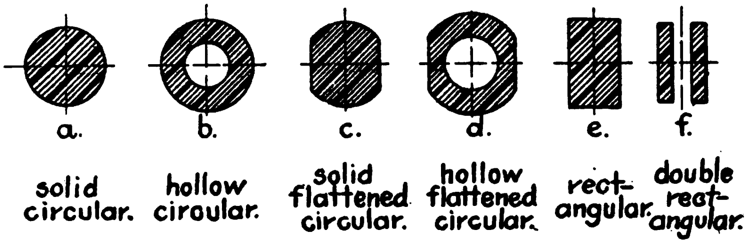
$\theta = \sin^{-1} \frac{24}{96} = \sin^{-1} \frac{1}{4} = 14° \, 29'$. Sec $\theta = 1.033$. Whence W = R sec θ = 158,600 × 1.033 = 163,800 lbs.

APPLICATION OF GORDON'S FORMULA.

C and N are given in the "specifications" as 80,000 lbs. and 8.5 respectively. $E = 30{,}000{,}000$ lbs. $\pi^2 = 10$. Thus we have

$$W = 163{,}800 = \frac{\dfrac{80{,}000}{8.5} \times A}{1 + \dfrac{80{,}000}{10 \times 30{,}000{,}000} \times \dfrac{96^2}{K^2}}.$$

A and K^2, our unknowns, are not independent. If made to depend on common variables (as d_1 and d_2 below) the equation is of high degree and not easily managed. It is best to solve simply by trial and error.



USUAL SECTIONS OF CONNECTING RODS.

FIG. 2.

Since we are confined to hollow sections (b or d in Fig. 2) because of the oil-tempering, our first trial will be a circular section of 8″ diameter with a 5″ bore hole. The diameter of the bore is usually from .5 to .6 of the outside diameter. We will calculate proper values for A and k from these dimensions and substitute in the right-hand member of the equation. The value of W found, will then be compared with the value required, 163,800 lbs.

Calling the diameter, area, and radius of gyration of the 8″ circle, d_1, A_1, and k_1, and those of the 5″ circle, d_2, A_2, and k_2, then $A = A_1 - A_2$.

Referring to the table of areas of circles on p. II, we find $A = 50.27 - 19.64 = 30.63$ sq. in.

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The value of k^2 in terms of d_1 and d_2 can be derived from the fact that the moment of inertia of the hollow section is equal to the moment of inertia of the solid 8″ circle — the moment of inertia of the 5″ circle. Expressed mathematically, $I = I_1 - I_2$, or $k^2A = k_1^2A_1 - k_2^2A_2$. Then $k^2 = \frac{k_1^2A_1 - k_2^2A_2}{A_1 - A_2}$.

But, $k_1 = \frac{d_1}{4}$, $k_2 = \frac{d_2}{4}$, $A_1 = \frac{\pi}{4} d_1^2$, and $A_2 = \frac{\pi}{4} d_2^2$. Making these substitutions,

$$k^2 = \frac{d_1^4 - d_2^4}{16(d_1^2 - d_2^2)} = \frac{d_1^2 + d_2^2}{16} = \frac{d_1^2}{16} + \frac{d_2^2}{16}.$$

A table of values of $\frac{d^2}{16}$ has been calculated and is given on p. II. From it we find $k^2 = 4.000 + 1.563 = 5.563$. Then,

$$W = \frac{\frac{80,000}{8.5} \times 30.63}{1 + \frac{80,000}{10 \times 30,000,000} \times \frac{96^2}{5.563}} = \frac{9,410 \times 30.63}{1 + \frac{2.4574}{5.563}} = \frac{288,200}{1.442} = 199,900 \text{ lbs.}$$

As this value exceeds 163,800 lbs. by a considerable amount we make a second trial reducing each diameter by ½″. Then:—

$d_1 = 7\frac{1}{2}''$, $d_2 = 4\frac{1}{2}''$, $A = 44.18 - 15.90 = 28.28$, $k^2 = 3.516 + 1.266 = 4.782$, and $W = \frac{9,410 \times 28.28}{1 + \frac{2.4574}{4.782}} = \frac{266,100}{1.514} = 175,800$ lbs.

This value is almost exactly two-thirds of the way from 199,900 to 163,800 lbs. Reducing again by ¼″ we may expect a very close result. Thus, $d_1 = 7\frac{1}{4}''$, $d^2 = 4\frac{1}{4}''$, $A = 41.28 - 14.19 = 27.09$, $k^2 = 3.285 + 1.129 = 4.414$, and $W = \frac{9,410 \times 27.09}{1 + \frac{2.4574}{4.414}} = \frac{254,900}{1.557} =$ 163,700 lbs., a solution as close as need be.

The diameter of rod and bore hole are, then, 7¼″ and 4¼″.

SECTION AT THE NECK.

The neck is assumed to be at about one-third the length of the rod from the cross-head end, say 30″. The true position of the neck will be determined later. The section of the rod at 30″ from the free end is theoretically the middle section of a rod 60″ long. The bore hole, 4¼″ in diameter, is of course retained and at first trial the outside diameter may be supposed to decrease about ½″. Then, $d_1 = 6\frac{3}{4}''$, $d^2 = 4\frac{1}{4}''$, $A = 21.59$, $K^2 = 3.977$, from which

$$W = \frac{9{,}410 \times 21.59}{1 + \dfrac{80{,}000}{10 \times 30{,}000{,}000} \times \dfrac{60^2}{3.977}} = \frac{203{,}200}{1.241} = 163{,}700 \text{ lbs., a}$$

very good result, so that no further trial is needed.

The value 203,200 lbs. is the strength of the rod as a tie rod, since it is the strength of the smallest section.

THE FORK.

The fork is faced at the sides to a thickness greater than the diameter at the neck by from 5 to 25%. In this case we may choose 8″ (A in Fig. 3). The width between the jaws is ⅛″ or ¼″ greater than the specified length of the cross-head pin. This allows a clearance, or room for a slight motion of the rod. The distance B (Fig. 3) is therefore 14¼″. The distance C must be such that the fork of the rod may clear by a large margin (1″ to 6″) the cap and bolts of the cross-head brasses in every position that the rod may take relative to the cross-head. It must be obtained from the drawing of the cross head, as shown in Fig. 4, or must be given in the data prepared by a previous designer. In this case C is 19″. The radius D is of course ½ B.

The thickness E must be such that the section of the fork is from 70 to 100% of the rod at the neck. Large rods require the larger amount as the tempering of the thick metal (over 2½″), is imperfect. It is supposed that much more than half the load may come on one side of the fork from defective alignment. Taking 95% in this case we have .95 × 21.63 = 20.55 sq. in. Since the width of the section is 8″ the average height must be $\frac{20.55}{8} = 2''.57$

$= 2\frac{9}{16}''$. If we take $E = 3''$ it will allow amply for the rounding down of the corners of the section when the outer surface of the fork is turned in the lathe, the rod being centered on its axis. (In Fig. 3 the dotted arcs, m and n, complete the circle traced out by the tool in cutting these surfaces.)

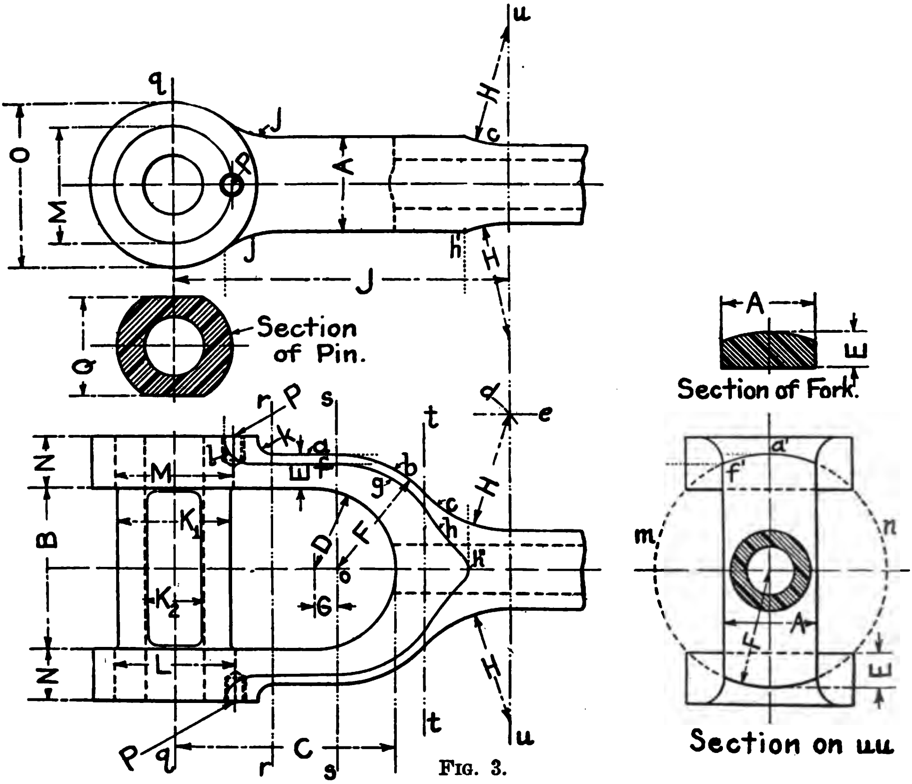


FIG. 3.

The radius F is equal to $D + E = 7\frac{1}{8}'' + 3'' = 10\frac{1}{8}''$. The longitudinal contour of the fork is continued to the right of the parallel portion (a) by running into the circular arc (b) of the same radius F. The center of this arc should be from $\frac{1}{2}''$ to $2\frac{1}{2}''$ to the right of the center of D. This strengthens the fork at the curved part, and it is still further strengthened where it joins the middle

length of the rod by rounding in with a large radius, H, about equal to F. In this case G = 2″ and H = 10″.

The distance J is measured on the accurate drawing of the rod. It is the true distance of the neck from the center of the cross-head pin. If it is less than the assumed 30″ the error is on the safe side and no recalculation is needed. In this case J = 28¾″.

CROSS-HEAD PIN.

The bearing length of the pin, 14″, has been specified to suit the design of the cross-head brasses. The diameter K_1 has been found by allowing a pressure of 1,200 lbs. per square inch of *projected area.* Thus 14 × 9¾ × 1,200 = 163,800 lbs. The bore hole K_2 lightens the pin, and enables it to be oil-tempered. The pin is of the same steel as the rod and is shrunk or forced into the eyes of the fork. A hole is then drilled and tapped half in the pin and half in the eye, as shown at P. A "screw plug" or "stud" is screwed tight in the hole and the head cut off flush with the surface. This plug prevents the pin from working loose and turning in the fork. For a small rod it may be as small as ½″ diameter × ¾″ long. For a large rod two plugs are used as here shown. These are 1½″ diameter × 2″ long.

The diameter L is slightly larger than K_1 to allow for a small fillet at the end of the bearing length of the pin. In large and medium rods this increase is ⅛″ and $\frac{1}{16}$″. Small rods do not require it. In addition M is greater than L by from $\frac{1}{32}$″ to $\frac{1}{16}$″. The small end L of the pin will then slip easily through the large bore M of the fork in the process of inserting the pin. In this case L = 9⅞″ and M = 9$\frac{15}{16}$″.

The eye of the fork is reinforced by making the depth N greater than E, and the diameter O greater than A. The area left in the plane qq after boring the hole M is equal to from 90% to 100% of the area of the neck. This rule is empirical. The exact strains cannot be calculated owing to the unknown bursting strains due to the pin. Choosing 95% we get (O — M) N = 20.55 sq. in. Assuming O = 14½″, N = 4½″.

The arc j is usually given a radius equal to A.

The arc k has a radius equal to or less than N — E. In this case it is 1¼″.

Most of the bearing length of the pin is faced at the two sides to a width Q somewhat less than K_1. The sides are useless as bearing surfaces since the force acting is always in line with the axis of the rod. By cutting them away a pocket is formed for oil to collect in, and to act as a kind of reservoir. A small length at each end, say ¼″, is left circular to help retain this oil.

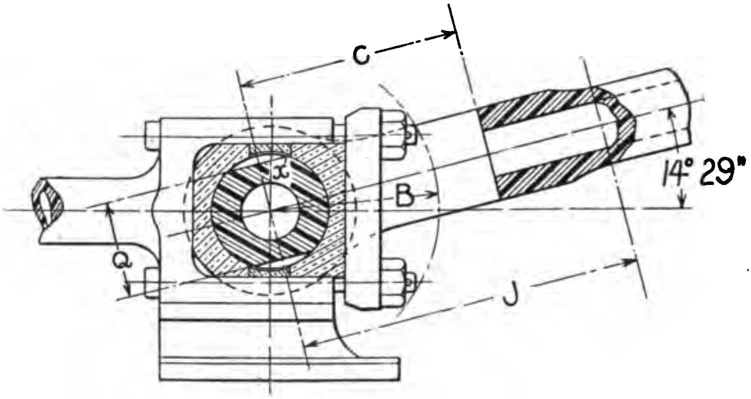


FIG. 4.

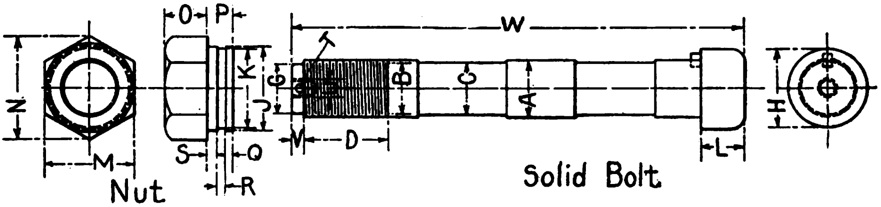
Fig. 4 is a sketch of the cross head with the fork end of the rod shown at the extreme angle of swing. The cross-head pin and brasses are sectioned and the near side of the fork removed. The width Q is so adjusted as to allow the edge, x, of the bearing surface of the pin to slightly overtravel the bearing surface of the brass it works against. All bearing surfaces whose action is *reciprocating* should be designed with this overtravel at the point of extreme displacement. It prevents a shoulder or step being formed in the brass by the wear not extending to the edge. Without going into the design of the cross-head brasses we shall assume that we know the distance Q and that it is in this case 8⅞″.

THE SIZE OF BOLTS, CRANK END.

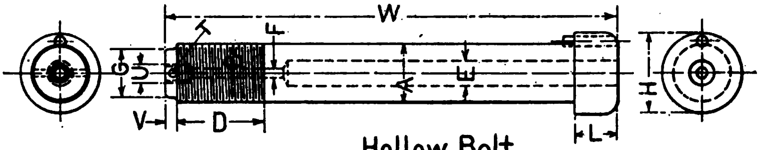
The general shape of the crank-pin brasses is seen in Fig. 6. The cap is held by two bolts, made of the same steel as the body of the rod. We imagine that ⅜ of the whole load on the rod may

come on one bolt through unequal tightening of the nuts. The bolt is never in compression, but its load fluctuates with each revolution of the engine from the designed tension to zero tension. A lower factor of safety can be used than for the rod itself in which the fluctuation is from tension to compression.

For torpedo boats and destroyers (thrust in connecting rod, 0 to 100,000 lbs.) the factor averages 5.8 in the designs of our naval vessels. For cruisers and battleships (thrust 130,000 to 245,000 lbs.) it averages 7.8. For gunboats (thrust 0 to 40,000 lbs.), 8.5.



Solid Bolt



Hollow Bolt

Fig. 5.

In the present case we will use a factor 7.5. Then for the area of the bolt at the weakest section we have:—

$$\text{Area} \times \frac{80{,}000}{7.5} = \tfrac{2}{3} \times 163{,}800. \quad \therefore \text{ area} = 10.22 \text{ sq. in.}$$

The two styles of bolts in general use are shown in Fig. 5. The solid bolt is turned down in two places so that the area remaining is about equal to the area of the threaded part at the bottom of the threads. The bolt thus reduced and made of fairly uniform strength along its length is stronger to resist sudden shocks than if left of full size. It is more yielding, more "resilient." By referring to the "Table of Standard Bolts and Nuts for the U. S. Navy," on p. I, and the "Table of Areas of Circles" on p. II, the dimensions, $A = 4''$, $C = 3\frac{5}{8}''$, will be found suitable for our case. The area at B is thus 10.61 sq. in., and at C it is 10.33 sq. in.

The second design accomplishes the purpose of making the bolt of uniform strength along its length by means of the bore hole E. The 4″ bolt has a section of 12.57 sq. in. or 2.34 sq. in. more than the 10.22 sq. in. required. A bore hole, E, of $1\frac{5}{8}''$ diameter will remove 2.07 sq. in., leaving the area at A equal to 10.50 sq. in.

For convenience in oil-tempering the bolt, the large bore hole E is usually continued by a small one, F, through to the end. If $F = \frac{5}{8}''$ the metal remaining at B will be $10.61 - .31 = 10.30$ sq. in.

Beyond the nut the bolt is turned to the diameter $G = C = 3\frac{5}{8}''$ and is extended far enough to allow a round split pin to be passed through the hole T. Split pins vary from $\frac{1}{8}''$ to $\frac{3}{4}''$ in diameter, depending on the size of the bolt. In this case the diameter is $\frac{1}{2}''$. The pin should bear against the face of the nut when in position.

The projecting end of the bolt is $\frac{3}{4}''$ long for the $\frac{1}{2}''$ split pin.

The length of the bolt and the positions of the turned down places cannot be determined until the brasses and cap have been designed. The threaded length of the bolt should exceed the depth of the nut by from $\frac{1}{2}''$ to 1″. Thus

$$D = P + O + 1'' = 5\frac{11}{16}''.$$

EYE-BOLT HOLES AND SET SCREWS.

At U a hole is drilled and tapped for a 1″ eye bolt. The depth of the hole is $1\frac{1}{2}''$. Medium-sized bolts are tapped for $\frac{3}{4}''$ eye bolts, $1\frac{1}{4}''$ deep. Set screws are fitted in the cap or in the brasses to hold the bolts after the nuts have been slackened and removed, until the eye bolts have been screwed into place and slings fitted. No groove is required for these set screws as they always bear at one place which can be flattened with a file. Small bolts require no set screws or eye-bolt holes.

Set screws are standard square-headed steel screws (see table on p. I). The head of the $\frac{3}{4}''$ set screw here used has a short diameter of $1\frac{1}{4}''$ and a long diameter of $1\frac{3}{4}''$. If necessary, it can be made much smaller than the standard. The point is turned down to $\frac{19}{32}''$ diameter. It projects $\frac{1}{4}''$ and is flat across the end.

BOLT HEADS AND NUTS.

The bolt heads are round and of less dimensions than the standard heads, as no twisting force is ever brought upon them. They are held from turning by dowel pins or "snugs." For the solid bolt, which has an excess of strength in the section just under the head, the snug is rectangular and is set into the metal of the *bolt*. In this case it is $\frac{5}{8}''$ square, set in $\frac{1}{2}''$ and projecting $\frac{1}{2}''$. For the hollow bolt it is a pin set in the metal of the *bolt head* by drilling completely through the head. For this bolt it may be $\frac{5}{8}''$ in diameter and $\frac{3}{4}''$ longer than the depth, L, of the bolt head.

The under surface of the bolt head is in compression. This area should equal the section of the bolt. The *whole* area of the bolt head is therefore $10.23 + 12.57 = 22.80$ sq. in. This will be given by a diameter, H, of $5\frac{1}{2}''$.

The thickness of the bolt head, L, is $\frac{3}{4}A = \frac{3}{4} \times 4'' = 3''$.

The nut is of wrought iron, case-hardened on the outside. The short and long diameters, M and N, are taken from the Table of Standards. They are $6\frac{1}{8}''$ and $7\frac{1}{16}''$ for a $4''$ bolt.

The bearing surface of the nut is the bottom of the nut. The bottom of the hexagonal portion stands clear of the metal by $\frac{1}{16}''$, when the nut is screwed home. The diameter J is a little larger than H, in this case $5\frac{3}{4}''$. The groove for the set screw should be $\frac{1}{16}''$ or $\frac{1}{8}''$ deep. Without the groove the burr raised by the pressure of the set screw would injure the fit of the nut. Thus we take $K = 5\frac{1}{2}''$. The height of the hexagonal part of the nut is $\frac{3}{4}$ of the diameter of the bolt. $O = 3''$. The height P is from $\frac{1}{2}$ to $\frac{2}{3}$ of O. The groove R is in this case $\frac{5}{8}''$ wide to allow for a $\frac{3}{4}''$ set screw. We will take $Q = \frac{1}{2}''$ and $S = \frac{9}{16}''$, thus making $P = 1\frac{11}{16}''$.

POSITION OF CENTER LINES OF BOLTS.

The inside edges of the bolts should clear the crank pin by from $\frac{1}{8}''$ to $\frac{3}{4}''$, according to the size of the crank pin. In this case $\frac{5}{8}''$ is sufficient. The distance from center of crank pin to center of bolt is therefore

$$\frac{\text{crank pin}}{2} + \frac{5}{8}'' + \frac{\text{bolt diameter}}{2} = \frac{14\frac{3}{4}''}{2} + \frac{5}{8}'' + 2'' = 10''.$$

THE STUB END OF THE MAIN FORGING.

The width of the "stub end" (parallel to the crank pin) is from 60% to 80% of the length of the brasses. The crank pin is 17″ long (see Specifications on p. 25). T, the corresponding length of the brasses (Fig. 6), is $16\frac{3}{4}''$, to allow the same side clearance, $\frac{1}{4}''$, as is allowed for the cross-head pin. Taking 80% we find $D = .80 \times 16''.75 = 13''.4 = 13\frac{1}{2}''$.

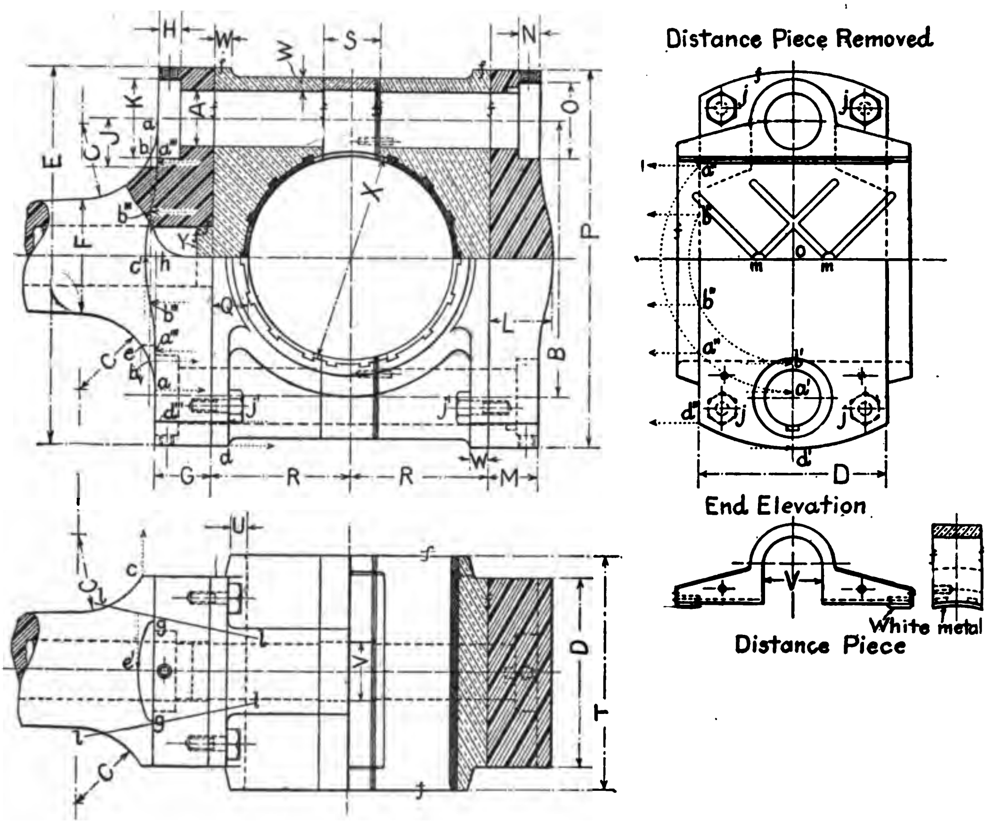


FIG. 6.

The distance of the butt from the center of the crank pin must now be determined. The thickness of the brass at Q is about $\frac{1}{3}$

$+a =$ [illegible]

$\frac{5}{6}$" [illegible]

the radius of the crank pin. Thus $Q = \frac{1}{6} \times 14\frac{3}{4}'' = 2\frac{1}{2}''$ nearly. The distance R is equal to $2\frac{1}{2}'' + 7\frac{3}{8}'' = 9\frac{7}{8}''$.

The width, E, of the stub end, is equal to the distance apart of the bolts + the long diameter of the nut + a small amount for clearance. In this case let $E = 20'' + 7\frac{1}{16}'' + \frac{3}{16}'' = 27\frac{1}{4}''$. This allows $\frac{3}{32}''$ at each end for the corners of the nuts to keep within the edges of the stub end.

The thickness G must allow sufficient metal for strength after recessing the seat for the nut. If G is made equal to the diameter of the bolt, (4″), the proportion will be a good one. The distance H is $\frac{1}{16}''$ less than P in Fig. 4. Thus $H = 1\frac{5}{8}''$.

The radius C is determined when the laying down of the design on the drawing board has reached this point. The arc must be tangent to the tapered section of the rod and also to the surface of the stub end at a point, a, at or within the center line of the bolt. It is usual to so adjust this radius as to make F, the diameter of the rod at the point of tangency, some integral number of inches and eighths, for convenience in the shop. If necessary, the diameter at the middle section, bb, in Fig. 1, is slightly increased to allow this. In this case, C may be made $5\frac{1}{2}''$ and F, $8\frac{1}{8}''$. The same radius, C, appears twice on the side elevation, the centers being on the same vertical line as the centers in the plan.

In addition to the recess for the circular part of the nut there is a recess to allow room for the corners of the hexagon as the nut turns. This recess is nearly a semicircle. Its diameter is slightly larger than the long diameter of the nut. In this case it is $7\frac{1}{8}''$, or $J = 3\frac{9}{16}''$.

The end surfaces of the stub end are cylindrical. In the end elevation they appear as arcs of a circle whose center is O.

THE CAP.

The cap is of oil-tempered nickel steel and is regarded as a beam supported at the ends and loaded with a uniformly distributed load. In other words, the bolts are not counted on to keep the ends of the cap from assuming a slight angle with their unstrained position. Undoubtedly they do resist such a tendency but the strengthening effect gained may be offset by the fact that the load is in reality distributed over the middle three-quarters of the length of the beam only. (The length of the beam is the distance between the centers of the bolts.)

The greatest bending moment is at the middle, therefore, and is equal to $\frac{Wl}{8} = \frac{163,800 \times 20}{8} = 409,500$ inch-lbs.

The moment of resistance of the rectangular section at the middle is:—$f_t \times \frac{bh^2}{6}$. (b, = the width, D, of the cap, and h = the thickness L.) Using the same factor of safety as for the rod, 8.5,

$$f_t, \text{ the safe working stress, } = \frac{80,000}{8.5} = 9,410 \text{ lbs.}$$

The equation for h, or L, becomes, therefore:—

$$409,500 = 9,410 \times \frac{13\frac{1}{2} \times h^2}{6}, \text{ or } h^2 = \frac{409,500}{9,410 \times 13.5} = 19.34,$$

whence h = 4″.4. We shall increase this to 4″.5.

The thickness of the cap at the ends, M, is about $\frac{3}{4}$ L. In this case $3\frac{1}{2}''$. For small caps, M = L.

Large caps have the bolt heads sunk in a distance N, a little less than H. For our cap, let N = $1\frac{1}{2}''$. The diameter of the recess is $\frac{1}{8}''$ larger than the diameter of the bolt head. Thus O = $5\frac{5}{8}''$. A recess receives the snug on the bolt.

P is equal to E. When the bolt heads are not recessed, P = distance apart of the bolts, + diameter of bolt head + $\frac{1}{8}''$.

The set screw which holds the bolt when the nut is slackened may set to press against the bolt head or the bolt itself, as is most convenient.

THE CRANK-PIN BRASSES.

The dimensions Q and R have been determined.

S, the distance taken up by the "distance piece" and tin liners, varies greatly with the individual designer. A good value for a $14\frac{3}{4}''$ pin is 4″. Of this amount $\frac{1}{4}''$ is taken up by the tin liners.

The two large brasses are alike, save for the boss Y on one, which fits into the bore hole of the rod. The thickness U of the

brasses at the overhang is about 10% of the length of the crank pin. In this case, we take it as 1½″ as the unsupported "lip" of the brasses is small. The outer surface of the lip is conical, the taper being 15° on each side.

Most of the other visible surfaces of the brasses are obtained by keeping the thickness as constant as possible. In this case a thickness, W, of 1″, is taken, at the places shown in Fig. 6.

Frequently tap bolts are used to fasten the "cap brass" to the cap and the "rod brass" to the stub end. Four are used for each, and are so placed as to utilize the waste material in the corners of the cap and rod. They are here shown at jj on the end view and j′j′ on the plan, and are 1″ in diameter. The holes in the cap and forging are drilled and tapped 1½″ deep.

Dowel pins projecting from the cap brass serve to keep the distance pieces and lines in place. Two pins are used for each distance piece. The pins may be ½″ diameter × 2″ long. The holes in the brass are ½″ × 1″ deep and are a "driving fit." The holes in the distance pieces are ½″ + 1½″ deep and are a "working fit."

The distance pieces are of horse-shoe shape, for convenience in removing them. The cap bolts need only be slackened back enough to free the distance pieces from the dowel pins. The diameter, V, of the concave semicircle is equal to the bolt diameter + 1/16″.

WHITE METAL LINING.

The brasses and the ends of the distance pieces are lined with white, or anti-friction, metal. The finished thickness ranges from 3/16″ to 7/16″ with the size of the rod. Taking it as ⅜″, the inside diameter, x, of the unlined brasses becomes 14¾″ + ¾″ = 15½″. In addition, many grooves, ¼″ deep × ½″ wide, with undercut edges, are spaced at equal angles around the circle. Two at least must be put in each distance piece. These grooves hold the white metal firmly in place.

The middle of the distance pieces and adjacent parts of the brasses are not lined, thus leaving a shallow cavity (⅜″ deep) in which oil collects and from which it is redistributed over the crank pin.

The white metal, when cast over the concave surfaces of the brasses, is made considerably thicker than the finished work. It is hammered down to a dense condition with a riveting hammer. The parts having been assembled and bolted together, the metal is bored out and scraped to an exact fit with the crank pin.

OIL HOLES AND DUCTS.

The provision for oiling the crank pin is usually given in detail on a separate drawing. Two copper tubes of ¾″ outside diameter, one running down each side of the rod, lead the oil from the crosshead end, where it is received drop by drop in a narrow funnel-shaped opening, to holes bored in the main forging along the lines ll. The holes extend nearly to the brass with the diameter of ¾″ and are continued through the brass and white metal with a smaller diameter.

From the openings on the surface of the white metal (see points m m on the end elevation, Fig. 6), oil ducts are cut by a grooving chisel leading in some pattern intended to spread the oil over the surface of the crank pin.

Sometimes the oil is introduced down the center line of the rod by a tube inside the bore hole. In such a case the hole through the brass and white metal is in the center of the boss Y and the oil ducts radiate from the point o in the end elevation.

ALTERNATIVE DESIGNS.

SOLID CIRCULAR MIDDLE SECTION.

This section is now little used in naval vessels, as it is not a favorable form for oil-tempering. It may be used for gunboats.

Since k, the radius of gyration of a disc about a diameter, is one-fourth the diameter, the formula can be written

$$W = \frac{\frac{C}{N} \times \frac{\pi}{4} d^2}{1 + \frac{16C}{\pi^2 E} \times \frac{l^2}{d^2}},$$

and solved by trial and error, but the application of the general formula, making $d_2 = o$, is just as simple.

FLATTENED CIRCULAR SECTIONS.

These sections (c and d of Fig. 2) are modifications of the sections a and b. The regular solution for the section a or b is first made including the size at the neck. Usually the section at the neck is not flattened at all and the middle section is flattened at the sides to a width equal to the outside diameter at the neck. (Of course the whole middle length is then faced to this width.) The flattening diminishes the area of the section, and to compensate, the outside diameter is increased. Graphical tables of the radii of gyration of flattened circular sections, solid and hollow, have been constructed and by their use the exact increase can be determined.

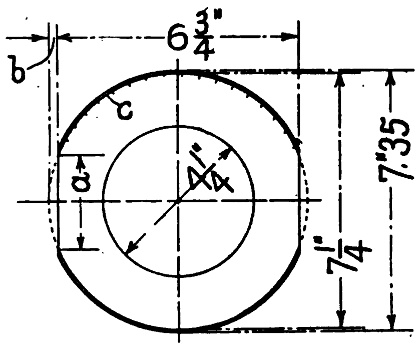


FIG. 7.

A fair solution, of which the error is on the safe side, is obtained by making the added area *equal* to that removed. Fig. 7 shows by this method the determination of the increase in the diameter of the rod of the Alabama at the middle, if it is desired to face the sides to the diameter at the neck. The area removed is approximately ⅔ × the chord a (which is by measurement 2⅝") × the altitude b, or ⅔ × 2⅝" × ¼" = .437 sq. in. By stepping along the arc c with the bow spacer set to ½" we find the arc equal to 8¾". The increase required is therefore $\frac{.437}{8.75}$ = ″05. The new outside diameter is, therefore, 7″35.

RECTANGULAR SECTION.

Values for the width and thickness of the section are assumed and the formula used to test them in the regular trial and error method. The values of t and w must be such that $w \overset{=}{<} 2t$. Since $A = w \times t$, and $K^2 = \frac{w^2}{12}$, the formula becomes:—

$$W = \frac{\frac{C}{N} \times t \times w}{1 + \frac{12C}{\pi^2 E} \times \frac{l^2}{w^2}}.$$

The use of it is illustrated in the following example:

For a torpedo boat, given W = 52,000 lbs., l = 40″, C = 80,000 lbs., N = 8.5, assume first that w = 3½″, t = 2″. Then W = 46,500 lbs. Assume next, w = 3¾″, t = 2. Then W = 51,700 lbs., a good solution.

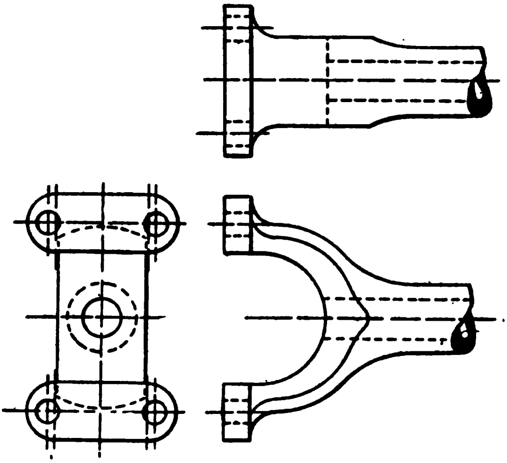


Fig. 8.

The thickness t is generally kept constant and the width w made to taper uniformly from the neck to the crank end. Assuming the neck to be at 13″ from the cross-head pin, we find that, if w = 3⅜″, and t = 2″, W will equal 53,400 lbs. This results in a taper of ⅜″ in 7″, an unusual amount, changed with advantage to ¾″ in one foot. This change will increase the width at the middle by $\frac{1}{16}$″, *i. e.* to $3\frac{13}{16}$″.

OPEN FORK, CROSS-HEAD END.

Large rods are sometimes designed with the jaws of the forked end fitted with small "marine ends," reduced copies of the crank end of the rod. In such cases the cross-head pin is shrunk in the cross head at the middle of its length, the two projecting ends forming end-journals for the brasses on the jaws. It is supposed that $\frac{2}{3}$ of the load may come on one jaw from irregularity in fitting, and the ends are designed for that load. The end of each jaw is forged out to the shape shown in Fig. 8, to give room for the bolts. White metal is often omitted from these brasses. The *nuts* of the cap bolts bear on the *caps,* not the fork.

"SOLID" CRANK END.

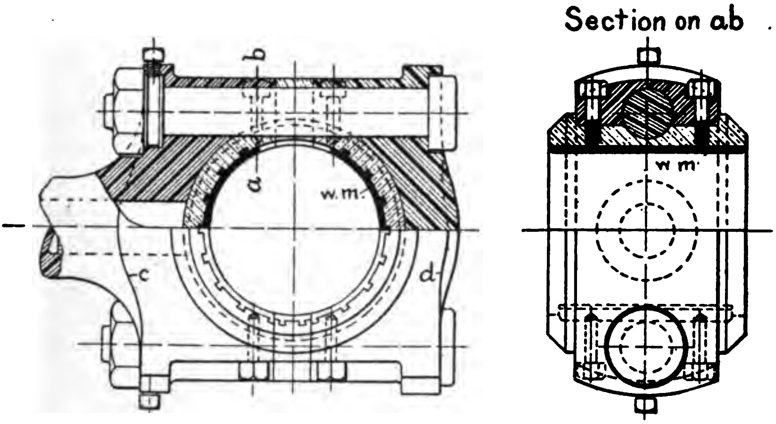


FIG. 9.

The crank end and cap are sometimes designed in the manner known as a "solid end." The brasses are reduced to thin semi-circular pieces held in the steel by "nipping bolts." The end of the main forging and the inner surface of the cap are concave semi-cylindrical surfaces instead of flat surfaces. To reduce the distance apart of the cap bolts, they are made to pass partly through the brasses. Fig. 9 shows this design. The section on a b is that through the nipping bolts.

CAP WITH ROUNDED ENDS.

The cap and the similar portion of the main forging are often finished off with semicircular ends, instead of ends formed by turning them on the center line of the rod. Thus in Fig. 6 we should have instead of the arc d″d′, whose center is o, a semicircle of diameter D, passing through the point d′. The two ends of the fork in Fig. 8 have been rounded in this manner.

BRASS CAP.

Small rods very frequently have a brass cap in lieu of a steel one. The cap in such cases takes the form of the steel cap in Fig. 9 combined in one piece with the cap brass. The rest of the design may follow either Fig. 6 or Fig. 9. In calculating the thickness of the brass cap, a low-working fiber stress is taken, its value being about half that for the steel. "Brass" caps are generally made of some form of bronze.

ORIGINAL CALCULATIONS AND SKETCHES FOR A CONNECTING ROD.

NOTE BOOK, SKETCHES, ETC.

Sets of specifications like those on p. 25 will be given out in the drawing room. From them each one will make his own design of a connecting rod for a naval vessel, following in a general way the methods given in the pages just preceding, but depending on his own judgment for a selection from among the alternatives mentioned for the different details.

He should from the beginning devote a special note book or part of a note book to keeping a permanent record of his calculations. The specifications are the first entry.

The calculations, as they are made, should be neatly entered in the book in conjunction with free-hand sketches of the details of the rod to which they refer. The sketches are of detached parts of the whole, like Figs. 3, 5 and 6, or of even smaller portions. Dimension letters are not used, but the values as determined by the calculations are recorded on the sketches which then become the basis from which the drawing of the rod is ultimately made.

ORDER OF RECORDING CALCULATIONS.

The following table gives the order in which the required dimensions are classified and recorded. It is, as far as possible, that in which they are calculated or determined:

In sketch books place dimensions on the left.

I *Specifications* (see p. 25).

II *Middle Length.*

1 Thrust on piston rod.

2 Sec $\left\{ \sin^{-1} \frac{\text{crank}}{\text{connecting rod}} \right\}$.

3 Thrust on connecting rod.

4 Factor of safety of connecting rod.

5 Type of cross-section, middle.

6 Dimensions of middle section.

7 A, K^2, and W for middle section.

8 Assumed position of neck.

9 Dimensions of section at the neck.

10 A, K^2, and W for section at the neck.

11 Strength of the rod as a tie rod.

12 Diameter of rod at cross-head end (to be filled later)

13 Corrected size at middle section (to be filled later).

14 True distance of neck from center of cross-head pin (to be filled later).

III *Fork End.*

15 Width of fork in plane of motion of rod.

16 Width between jaws of fork.

17 Area of fork required.

18 Thickness of fork at center.

19 Distance between centers for inside and outside circular contours of fork.

20 Radius for rounding in at the neck.

21 Diameter of cross-head pin within top eye of fork.

22 Diameter of cross-head pin within bottom eye of fork.

23 No. and size of screw plugs for fastening cross-head pin in eyes of fork.

24 Outside diameter of eye of fork.
25 Depth or thickness of eye of fork.
26 Radius for rounding in side lines of fork and outside diameter of eye of fork.

IV *Cap Bolts, Etc.*

27 Factor of safety of cap bolts.
28 Area of cap bolts.
29 Diameter and area of turned-down part of bolt.
or
Diameter and area of bore hole in bolt.
30 Diameter and area of small continuation of bore hole.
31 Diameter of bolt beyond nut.
32 Length of bolt beyond nut.
33 Diameter of hole for split pin.
34 Threaded length of bolt (filled later on).
35 Total length of bolt under head (filled later on).
36 Position of turned-down places (filled later on).
or
Depth of large bore hole in bolt (filled later on).
37 Diameter and depth of eye-bolt holes.
38 Diameter of bolt head.
39 Depth or thickness of bolt head.
40 Dimensions of snug.
41 Short and long diameters of nut.
42 Depth of hexagonal part of nut.
43 Depth of cylindrical part of nut.
44 Width and depth of groove on nut.
45 Width of collars on each side of groove.
46 Size of set screws.
47 Size of heads of set screws, if not standard.
48 Diameter of point of set screws.
49 Length of point of set screws.
50 Length of set screws under head (to be filled later).
51 Distance apart of bolts.

V *Stub End and Cap.*

52 Width of stub end parallel to crank pin.
53 Width of stub end perpendicular to crank pin.

54 Thickness of stub end parallel to center line of rod.
55 Depth of recess for cylindrical part of nut.
56 Diameter of recess for hexagonal part of nut.
57 Radius of arc of enlargement by which middle length joins stub end.
58 Thickness of cap at center.
59 Thickness of cap at ends.
60 Depth of recess for bolt heads.
61 Diameter of recess for bolt heads.
62 Length of cap over all, perpendicular to crank pin.

VI *Brasses, Etc.*

63 Length of crank pin covered by brasses.
64 Length of overhang, or lip of brasses.
65 Thickness of brasses at overhang.
66 Thickness of brasses between crank pin and stub end.
67 Distance from center of crank pin to stub end.
68 Thickness of distance pieces.
69 Thickness of tin liners.
70 Thickness of brasses over bolt.
71 Thickness of flanges.
72 Size and position of tap bolts fastening brasses to forgings.
73 Size and position of dowel pins for aligning distance pieces.
74 Width of horse-shoe opening in distance pieces.
75 Thickness of white metal lining.
76 Depth and width of undercut grooves for holding white metal.
77 Number and angular position of grooves.
78 Dimensions of cavity where white metal is not placed.
79 Diameter and direction of oil holes through main forging.
80 Diameter of oil holes through brass and white metal.
81 Pattern and size of oil ducts or grooves.

The different groups into which these dimensions have been collected should have their own sketch or sketches to illustrate them and to show the exact details.

Some of the dimensions are here entered in advance of their natural order of determination. Such are left blank for the moment. A few are determined only on the accurate drawing of the connecting rod. With the note book otherwise complete proceed to lay down the design on the drawing board, but finally return to the note book and fill in those dimensions here marked "to be filled later."

THE WORKING DRAWING OF THE CONNECTING ROD.

ARRANGEMENT OF VIEWS AND SECTIONS.

The Figures in the preceding pages illustrate special points and are not models to be followed in the drawing.

The connecting rod should be shown with *all parts assembled.*

For a *vertical* engine the *plan,* in an architectural sense, would be an end view of the rod from the fork end. Practically the rod is considered as lying on one side, and that view in which the cross-head and crank pins show as circles, is taken as the "plan." The (other) side elevation is placed under the "plan." Contrary to the general rule it is advisable to put the end view or elevation (looking on the *crank* end) on a line with the "plan," not with the "side elevation." This arrangement is shown on a small scale in Fig. 1. It makes a compact drawing, as the end view is narrowest in the direction parallel to the crank pin and it leaves a space convenient for the legend.

When necessary to show the interior structure, parts of the plan and elevations are made "in section," as in Figs. 6 and 9. (In Fig. 6 the bolts are absent, as is often the case in *sketches.* In the *drawing* they should be in place, as in Fig. 9.)

Often small additional sections are drawn, without showing any part beyond the plane of the section itself, like the sections of the cross-head pin and the fork in Fig. 3. These are intended to show clearly to the eye, the strength of the metal at those places.

THE SCALE OF THE DRAWING.

The scale depends on the size of drawing space to be devoted to the rod. The cutting, border, and working lines are first laid

down to the sizes specified in the drawing room. The *actual* distances, c and d, on the connecting rod (see Fig. 1) are computed and added together and the result compared with the horizontal clear space, e, between the two side-working lines. Such a scale is chosen as will make the *apparent size* of c + d a little less than e. The distance left over must provide first for a good separation between plan and end elevation, and then, if still sufficient, for additional space at the ends like f and g in Fig. 1.

CENTER LINES, CIRCLES, ETC.

The vertical center line through the cross-head pin is first laid out by measuring from the left-hand working line. From it is measured the center line of the crank pin. The vertical center line of the end view is measured from the right-hand working line. The horizontal center line of the plan is measured down from the top working line. The center line of the side elevation can be put in by eye alone unless the vertical height available is scanty. The center lines of the bolts are then drawn. The circles representing the cross-head and crank pins on the plan, the dotted circles for the diameters of the rod on the end view, the arcs for the ends of the cap and the full and dotted circles for the diameters of bolt heads and bolts are then drawn. The circles for the contour of the fork inside and out follow. As a general rule, circles are drawn first. Very often by projecting tangent lines from these circles from one view to another double measurements of the same distances are saved.

The points a a and b b are now established. Their horizontal distances from the cross-head pin are measured off, and their vertical positions projected from the end view. The rod as finally designed may not follow these points exactly but the contour must include them.

All the fundamental dimensions so far drawn should be verified and altered until absolutely correct.

The details of the ends of the rod are now taken up in much the same order as that of performing the calculation.

GEOMETRICAL LINES.

Some of the lines on the drawing are obtained, not by measured distances but by a *geometrical construction.* Thus, in the plan in Fig. 6, the line marked d‴ is derived by projection from the point d″ in the end view.

Many of these geometrical lines are of one class and represent the appearance of the edge or intersection of a surface of revolution and a plane parallel to its axis. Such curves are a‴ b‴ c′ in Fig. 6, c and d in Fig. 9, an f g h in Fig. 3.

The method by which points are found on such a curve is illustrated in Fig. 6, by finding the points b‴. The "surface of revolution" is in this case the "flared" or "bell-shaped" surface by which the middle length of the rod is enlarged to join the stub end. The plane is the flat side of the stub end. The contour of the flared part is formed, both on the plan and on the side elevation, by arcs of the *same* radius C, whose centers all lie on the *same* vertical line.

One point on the edge, c′, can be taken at once by projecting up from c on the side elevation. If any plane is passed perpendicular to the axis of the flared surface at a point between c′ and h, it will cut out a circle. b b is such a plane and it appears as a line on the plan, while on the end elevation the circle it cuts out is seen in its true shape. A part of the circle is the dotted arc b′ b″ b″, whose center is o, of which the point b′ is taken by projection from b on the plan. The two points, b″, lie both on this circle and on the flat edge of the stub end. Project from b″ to the plane b b on the plan. The points b‴ so marked are the same two points and are therefore on the required edge between the two given surfaces.

The plane h a is the limiting plane to the right, the plane to which the bell-shaped surface becomes tangent. In the same way therefore, a, the point of tangency of the arc of radius C, gives rise to the arc a′ a″ a″, and the points a″ give the points a‴ on the line h a. Outside of the points a‴ the curve is straight.

If the dimensions F and D are more nearly equal than in Fig. 6, the curve becomes a more marked one, and it may be well to

find more points than those here found, a‴, b‴ and c′. When enough points have been found, by passing new perpendicular planes, the curve can be drawn in with French curves, or arcs of circles found to approximate to it.

When the curve is as flat as in this case, since nothing depends on the accurate drawing of it, determine only the points a‴ and c′. Pass through them three arcs of circles tangent to each other, finding the centers by trial. The center of the middle one will be on the center line of the figure to the right of c′, the centers of the others to the left on horizontal lines through the points a.

The curve c in Fig. 9 is like that just described.

The curve d (Fig. 9) arises when the entire surface of the cap visible on the end view is turned on the lathe with the rod centered on end centers. Both c and d can be represented by three arcs of circles.

The curve f g h in Fig. 3 is of the same kind. The sectional view in Fig. 3 is not needed on the completed drawing, but some portions of it must be drawn in pencil for use in finding the curve f g h. A convenient place for it is at x in Fig. 1.

In Fig. 3 the outer surface of the fork between the planes r r and s s is cylindrical, and in consequence the edge f is straight and is given by projecting from the corner f′ in the sectional view. Between the planes s s and t t the surface is truly spherical. The edge g is therefore a circular arc whose center is O. Between t t and u u the surface is bell-shaped. The point h″ is projected down from h′ on the plan. Intermediate points can be found by passing planes between t t and h″, but the edge h is usually represented by three circular arcs tangent to each other and to the arc g, the middle arc passing through h″.

The line f (Fig. 3) is continued to the left by the line l, which meets the upper surface of the fork at a point projected from the plan, as shown. The exact drawing of the curve is immaterial. It is usually drawn a circular arc and need not continue quite to the upper surface of the fork (see Fig. 1). The line represents in reality the edge produced by the hand filing necessary to finish neatly the cutting out of the corners at j.

In Fig. 6 the recess cut in the stub end for the corners of the nut to turn in, is almost a semicircle. The line of the edge appears on the side elevation approximately as the half ellipse g e′ g, in which the major axis g g is equal to 2 × J (on the plan) and the point e′, the extremity of the minor axis, is projected from e on the plan. Any approximate method of drawing the ellipse by circular arcs will be sufficiently accurate.

MARKING FINISHED SURFACES.

A working drawing should show definitely which surfaces are machined and which are left in a comparatively rough condition ("rough forged" or "cast" surfaces). The general rule is to mark every planed, turned or slotted surface with an "f" (meaning "finished") on that part of the drawing where it is seen *on edge* and is represented by a *line*. The f is put right on the line it refers to. The letter "b" or the word "bore" is used to distinguish holes bored or drilled from those made by a core in the process of casting. It is put after the dimension figures giving the diameter of the bore hole thus $|\leftarrow \frac{3}{4}''\text{b} \rightarrow|$.

In most connecting rods, the brasses, which are castings, alone have unfinished surfaces, and it is customary therefore to put on the drawing in a prominent place a note to this effect "All surfaces finished except on brasses, which are finished where marked." The brasses, in Fig. 6, have been marked in conformity with this practice to show where to put the "f's."

NOTES ON THE DESIGN OF CRANK SHAFT, CROSS-HEAD PINS, ETC.

CRANK SHAFT.

In calculating the dimensions of a crank shaft for strength, the theoretical aggregate mean effective pressure is found and the actual pressure expected is determined by comparison with the results given by the trials of engines of the same type, pressure, and rate of expansion. In other words, a value for the efficiency of the valve gear, or the percentage of the actual to the theoretical mean pressure must be obtained (*Seaton,* p. 123).

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This efficiency is obtained as follows: Let r = total rate of expansion $= \dfrac{\text{area of L. P. cylinder}}{\text{area of H. P. cylinder} \times \text{cut-off in H. P. cylinder}}$.

Then the total mean pressure P_m is found from: $P_m = p_1 \times \dfrac{1 + \log_e r}{r}$; or, by using table, *Seaton*, p. 109: $P_e = P_m - p_o$.

This calculated value of P_e is compared with the actual value found from the trial trip, and the efficiency is found from:

$$x = \frac{\text{actual}}{\text{calculated}} P_e.$$

BEARINGS.

Pressure per Sq. In. of Projected Area with Regard to Heating:

Seaton:—The bearing surface of crank pins must be such that the pressure per sq. in. does not exceed 500 lbs., and, in the case of merchant ships where room will permit of larger pins, 400 lbs. should not be exceeded. When the brass is recessed so that it bears only on parts of the shaft, the actual bearing surface should not be exposed to more than 600 lbs. per sq. in. Main bearings of screw engines, when room admits, should be of such a size that the pressure does not exceed 200 lbs., measuring the whole of the bearing, or 300 lbs. on the actual bearing surface.

It is found that the channel ways take up about 20% of the bearing surface, leaving about 80% as the actual bearing surface.

The length of crank pins is from 1 to 1½ times the diameter of the pins. Vertical engines usually have space for a length of bearing equal to 1½ diameters. The length of each main journal is from 1 to 1½ times the diameter of the shaft. Vertical engines have usually a length of bearing equal to 1½ diameters.

Foley's Engineers' Reference Book: "No hard and fast rule can be laid down. The engineer must be guided by circumstances and experience. The velocity with which the surfaces move on each other must be considered, and also whether the direction of the pressure alternates. In the case of a cross-head pin, the motion is not only slow, but the pressure alternates as well; hence the high pressure permissible. The following may be a rough guide:

Cross-head pins, 750 lbs.; not to exceed 1,000 lbs. Crank pins, 300 lbs.; not to exceed 500 lbs. Main bearings, 250 lbs.; not to exceed 500 lbs."

The *Bureau of Steam Engineering* uses the following formulæ as guides:—

For crank-shaft journals: $p \sqrt{v} < 8{,}000$ (1). Where $p =$ pressure per sq. in. and must not exceed 500 lbs.; $v =$ velocity of surface in ft. per min.

For crank pins: $p \sqrt{v} < 15{,}000$ (2) and p must not exceed 700 lbs.

HOLLOW SHAFTS AND PINS.

The diameter of the equivalent hollow shaft may be calculated by the method of *Unwin*, p. 221: $d^3 = \dfrac{d_1^4 - d_2^4}{d_1}$.

Foley's practice is:

When dia. of hole	= .4	of d add	1%	to dia. of solid shaft.
"	= .5	"	2	"
"	= .6	"	5	"
"	= .7	"	10	"

This brings very nearly the same result as solving the equation above, which must be done by assuming some ratio between the diameters of shaft and bore hole or by assuming one diameter.

The *Bureau of Steam Engineering* makes the diameter of the crank pins one inch larger than that of the shaft journals. The length of the pins is calculated so that a certain pressure, about 400 lbs. per sq. in., is not exceeded; this length being generally 1¼ times the diameters of the pins.

CALCULATIONS.

The engine for which the following calculations are made is that of the U. S. S. "Raleigh."

Data: Vertical, inverted cylinder, direct-acting, triple-expansion, twin-screw engines. Cylinder diameters, 36″, 53″, 80⅝″.

Stroke, 33″. Revolutions per minute, 164. Boiler pressure 160 lbs., by gauge. 1st receiver pressure, absolute, 60 lbs. (assumed). Vacuum, 24″. Cut-off in H. P. cylinder, .7 stroke.

$$\frac{\text{Max. T. M.}}{\text{Mean}} = 1.5\text{; dist. between bearings, } 53''. \quad f = 9{,}000.$$

CRANK SHAFT.

The diameter and length of the main journals are computed for strength by two methods, and then, as a check, are calculated with regard to heating.

For strength (1), *Seaton*:—

$$r = \frac{\text{area L. P. cyl.}}{\text{area H. P. cyl.} \times \text{cut-off H. P. cyl.}} = \frac{5105.4}{1017.9 \times .7} = 7.$$

From the table, $\frac{p_m}{p_1} = .421$, for a value of 7.

$\therefore\ P_m = (160 + 15)\ .421 = 73.675$ lbs.

$\therefore$ Calculated $P_e = 73.675 - 3 = 70.675$ lbs., and expected $P_e = 70.675 \times .50 = 35.34$ lbs.

For one set of engines:—

$$\text{I. H. P.} = \frac{\text{PLAN}}{33{,}000} = \frac{35.34 \times 33 \times 5105.4 \times 164 \times 2}{12 \times 33{,}000} = 4931.6$$

$$\text{Mean T. M. of after engine} = \frac{\text{I. H. P.}}{\text{Rev.}} \times 63{,}000.$$

$$\text{Maximum T. M.} = \frac{4931.6 \times 63{,}000 \times 1.5}{164} = 2{,}841{,}700,$$

and $\overline{\text{Max. T. M.}}^2 = T^2 = 8{,}075{,}350{,}000{,}000$ (by logs).

M, the bending moment, $= \frac{W\,l}{8}$, where W = maximum turning force $= \frac{\text{Max. T. M.}}{\text{crank}} = \frac{2{,}841{,}700}{16.5} = 172{,}220$ and l = distance between bearings = 53″.

$\therefore M = \dfrac{W l}{8} = \dfrac{172{,}220 \times 53}{8} = 1{,}141{,}000$, and $M^2 = 1{,}301{,}900{,}000{,}000$ (by logs). $T_e =$ equivalent T. M. $= M + \sqrt{M^2 + T^2}$ $= 4{,}203{,}200$ (*Seaton,* p. 181).

$\therefore$ Dia. of shaft $= d =$

$$\sqrt[3]{\frac{T_e \times 5.1}{f}} = \sqrt[3]{\frac{4{,}203{,}200 \times 5.1}{9{,}000}} = 13''35.$$

By the method of *Unwin,* p. 267: $d = K \times \sqrt[3]{\dfrac{\text{I. H. P.}}{\text{Revs.}}}$.
K is obtained from

$$K = C\sqrt[3]{\frac{63{,}024 \times \pi}{16 \times f}}$$

$$C = \text{about } 4.$$

The value of K used by the Bureau of Steam Engineering is 4.3; hence $d = 4.3\sqrt[3]{\dfrac{4931.6}{164}} = 13''37$.

We have, then, by the two methods:—

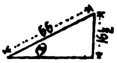
$$\left.\begin{array}{l} \textit{Seaton} = 13''35 \\ \textit{Unwin} = 13''37 \end{array}\right\} \text{average} = 13''36.$$

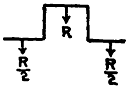
This is for a solid shaft. The shaft in question is to have a 6″ axial hole. Referring to *Foley's* practice, as above:

Dia. of hole $= \dfrac{6}{13.36}\,\% = 45\%$ of dia., nearly. By interpolating in the table we get 1.5% increase for $d_2 = .45\,d_1$. Then $13.36 \times 1.015 = 13''56$, or dia. of shaft, by calculation for strength, $= 13\tfrac{9}{16}''$.

LENGTH OF MAIN JOURNALS.

In order to obtain this, we must find first the thrust R on the connecting rod. This is calculated for the H. P. cylinder.





Effective load on H. P. piston $= P =$ area $\times$ ($p_1 - p_r$, or initial abs. press. — rec. press.) $= 1017.9 \times (175 - 60) = 117{,}058.5$. Then $R = P \sec (\sin^{-1} \tfrac{1}{4}) = 120{,}900$ lbs.

Then $\frac{R}{2}$, the thrust on each bearing, $= \frac{120{,}900}{2} = 60{,}450$ lbs.

From notes on "Bearings," p. 53, allowing a pressure of 190 lbs. per sq. in. of projected area, we have, for one bearing:—

$$l \times d \times 190 = 60{,}450$$
$$l \times 13.56 \times 190 = 60{,}450, \text{ and:—}$$

Length of bearing $= l = 23''.46$ or $23\frac{1}{2}''$.

The projected area required is $\frac{60{,}450}{190} = l \times d = 23.46 \times 13.56 = 318.13$. From this, $\frac{l}{d} = \frac{23.46}{13.56} = 1.73$, which is too large, as there is seldom fore and aft length allowed for such a coefficient. Therefore, the dimensions obtained must be changed as follows:

$$l = 1.5\,d \quad \therefore l \times d = 1.5\,d^2 = \text{projected area.}$$
$$\therefore 1.5\,d^2 = 318.13 \text{ and } d = 14.56, \text{ or } 14\tfrac{1}{2}''.$$
$$\therefore l = 14.56 \times 1.5 = 21.75, \text{ or } 21\tfrac{3}{4}''.$$

The projected area is, then, $14.5 \times 21.75 = 315.375$ sq. in. The pressure per sq. in. in this area $= \frac{60{,}450}{315.375} = 191.68$ lbs., which is within the limit of 200 lbs. The area of the actual bearing surface $= 315.375 \times .8 = 252.3$ sq. in., and the pressure per sq. in. on this area $= \frac{60{,}450}{252.3} = 239.6$ lbs., which is within the limit of 300 lbs.

The Bureau of Steam Engineering's check for speed:—

$$\text{From equation (1), velocity} = v = \frac{\pi d \times 164}{12} =$$

$$\frac{3.14 \times 14.5 \times 164}{12} \quad \therefore \sqrt{v} = 24.95 \text{ ft.}$$

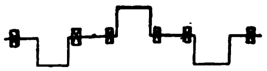
p, the pressure per sq. in. of projected area (from above), = 191.68 lbs. Then $p\sqrt{v} = 191.68 \times 24.95 = 4782.7$, which is within the limit of 7,500.

For the dimensions of the main bearings, we have, then:—

Diameter = 14½″; length = 21¾″.

CRANK PINS.

All crank pins are made of the same dimensions as the after pin, which is calculated for bending moment. To determine the bending moment on the after pin, consider the turning force of the H. P. and I. P. engines to be transmitted directly back to the forward end of the L. P. pin; or, what is the same thing, consider two-thirds of the turning force of the three engines transmitted to this point. Then, the pressure or load transmitted to the forward end of the L. P. pin is equal to two-thirds of the maximum twisting moment, divided by the length of the crank.



Let $\dfrac{\text{Max. T. M.}}{\text{crank}}$ or $\dfrac{T}{\text{crank}} = K$ and length of pin $= l$.

Then, $K = \dfrac{2{,}841{,}700}{16.5} = 172{,}240$. Since the transmitted force acts on the forward end of the pin, its bending moment $= \frac{2}{3} K\,l$. In addition to this load on the forward end, the pin takes the bending moment of its own engine, which is $\dfrac{\frac{K}{3} \times l}{8}$, since the pin is supported and loaded uniformly with a load $= \dfrac{K}{3}$. Hence, the total bending moment on the pin is:—

$$2\,\frac{K\,l}{3} + \frac{K\,l}{24} = \frac{17}{24}\,K\,l.$$

Substitute this bending moment in the formula for a shaft subjected to bending, *Unwin,* p. 200, $P\,l = \dfrac{f\,d^3}{5.1}$ where $P\,l$ = total

bending moment. Then $\frac{17}{24} K l = \frac{f d^3}{5.1}$.

$$\therefore d = \sqrt{\frac{17}{24} K \times \frac{l}{d} \times \frac{5.1}{9{,}000}}.$$

From notes on "Bearings," p. 53, $\frac{l}{d}$ is from 1 to 1.5. Take it as 1.25, since naval engines seldom have enough fore and aft space for the ratio 1.5. Then,

$$d = \sqrt{\frac{17 \times 172{,}240 \times 1.25 \times 5.1}{24 \times 9{,}000}} = 9.296 = 9\tfrac{5}{16}''.$$

Then, $l = 1.25\, d = 1.25 \times 9.296 = 11.62 = 11\tfrac{5}{8}''$.

These dimensions are for solid pins. For hollow pins apply the table given above.

Dia. of hole $= \frac{6}{9.296}\% = 65\%$ of d and 9.296×1.075 (interpolating in table) $= 9''.993 = 10''$, = diam. of hollow pin.

Also, length of hollow pin $= 10 \times 1.25 = 12.5 = 12\tfrac{1}{2}''$.

These are the dimensions of the pins for strength.

Check for Heating.—Pressure per sq. in. of projected area must not exceed 500 lbs. Assume a pressure of 450 lbs. per sq. in., and we have:—

Projected area of pin $= \frac{\text{load or thrust R on pin}}{450} = \frac{120{,}900}{450} =$ 268.67 sq. in.

Referring to the calculated dimensions for the crank pins, $10'' \times 12\tfrac{1}{2}''$, the projected area = 125 sq. in. This area, therefore, though ample for strength, is too small to prevent heating, for which an area of 268.67 sq. in. is required. Therefore, the dimensions of the pin must be increased:—

Then $(d \times 1.25)\, d = 268.67$. $\therefore d = 14''.66$ or $14\tfrac{3}{4}''$ and $l = 1.25\, d = 18''.44$ or $18\tfrac{1}{2}''$; and the projected area $= 14\tfrac{3}{4} \times 18\tfrac{1}{2} =$ 272.875 sq. in. The pressure per sq. in. on this area $= \frac{120{,}900}{272.875} =$

443 lbs. The actual projected area of bearing surface is, then:— $272.875 \times .8 = 218.30$ sq. in.; and the pressure per sq. in. on this area $= \dfrac{120{,}900}{218.30} = 553.8$ lbs., which is inside the limit of 600 lbs.

Bureau of Steam Engineering's check for speed:—From notes on "Bearings," Equation (2):—

$p\sqrt{v} < 15{,}000$, p not to exceed 700 lbs.

$$\sqrt{v} = \sqrt{\frac{\pi \times d \times 164}{12}} = \sqrt{\frac{3.14 \times 14.75 \times 164}{12}} = 25.16 \text{ ft.},$$

the slight difference in v, owing to the angularity of the connecting rod, being left out of the calculation.

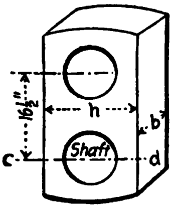
Also, from above, $p = 443$ lbs.; then, $p\sqrt{v} = 443 \times 25.16 = 11{,}148$, which is within the limit of 15,000.

It is seen, then, that the dimensions found fulfill all conditions, the changes in diameter keeping the hollow pin amply large for strength.

The dimensions of the crank pins are, then:—Diameter $= 14\frac{3}{4}''$; length $= 18\frac{1}{2}''$.

Bureau of Steam Engineering's practice:—

Dia. of shaft $= 14\frac{1}{2}''$. Then dia. of pins $= 15\frac{1}{2}''$. Length of pins $= 1.25 \times 15''.5 = 19''.375$ or $19\frac{3}{8}$. Bearing pressure per sq. in. of projected area $= \dfrac{120{,}900}{l \times d} = \dfrac{120{,}900}{15\frac{1}{2} \times 19\frac{3}{8}} = 402.7$ lbs., which is within the limit of 500 lbs. per sq. in. These dimensions are slightly larger than those found by the above calculations, and are on the safe side for both strength and heating.



AFTER ARM OF AFTER CRANK.

Seaton's method, p. 190: The breadth b is assumed $= 8\frac{1}{2}''$. Since the leverage is at the pin, the greatest thickness is required at *c d*. Calculate for the greatest thickness of the crank arm at *c d*, and make the sides parallel.

The bending moment at *c d* is the entire twisting moment on the crank shaft at the after bearing, and $\frac{b\,h^2}{6}$ = modulus of section (rectangular). Therefore $\frac{b\,h^2}{6}\,f = \text{Max. T. M.}$

$$\therefore h = \sqrt{\frac{\text{Max. T. M.} \times 6}{b \times f}} \text{ and } b = \frac{\text{Max. T. M.} \times 6}{h^2 \times f}.$$

Substituting the values of Max. T. M. and f (9,000 lbs.),

$$h = \sqrt{\frac{2{,}841{,}700 \times 6}{8.5 \times 9{,}000}} = 14''.93 \text{ or } 15''.$$

By this method, then, $b = 8''.5$ and $h = 15''$.

Bureau of Steam Engineering's Method:—The check used by the Bureau is to make the width of the crank arm across the face, h, about two inches more than the diameter of the shaft, and then compute the value of the thickness b, by the formula above for b, or by using the formula in *Unwin,* p. 225, $b\,h^2 = c\,d^3$, giving *c* a value of .9 for steel.

1st Method:— Dia. of shaft $= 14\frac{1}{2}''$. $\therefore h = 16\frac{1}{2}''$, and

$$b = \frac{281{,}700 \times 6}{(16\frac{1}{2})^2 \times 9{,}000} = 6''.96.$$

2d Method:— $b\,h^2 = c\,d^3$. $\therefore b = \frac{.9 \times (14\frac{1}{2})^3}{(16\frac{1}{2})^2} = 10''.08.$

The average of these two methods gives $8\frac{1}{2}''$, the thickness assumed for *Seaton's* method. The dimensions of the after arm of the after crank are then, $b = 8\frac{1}{2}''$; $h = 15''$.

CROSS-HEAD JOURNALS.

These dimensions are calculated in accordance with the methods of *Unwin,* Part I, p. 194.

Assume that two-thirds of the thrust R of the connecting rod may come on each journal. This allows for inaccurate adjustment and irregular wear.

$$\tfrac{2}{3}\,R = P = \tfrac{2}{3} \times 120{,}900 = 80{,}600 \text{ lbs.}$$

$\frac{Pl}{2} = f \frac{\pi}{32} \frac{d_1^4 - d_2^4}{d_1}$. In this case, let $\frac{l}{d} = 1.1$, and the axial hole $d_2 = 4''$; $f = 9{,}000$ for steel. The ratio of $\frac{l}{d}$ is taken small to prevent too great a spread of the jaws of the connecting rod, thus making a lighter rod. Substituting,

$$\frac{1.1 \times 80{,}600\, d_1}{2} = 9{,}000 \times \tfrac{22}{7} \times \tfrac{1}{32} \times \frac{d_1^4 - (4)^4}{d_1}.$$

Solving for d_1 we have $d_1 = 7''.43$ or $7\frac{1}{2}''$,

Then, $l = 1.1\, d = 1.1 \times 7''.5 = 8''.25$ or $8\frac{1}{4}''$.

This gives dimensions for strength. The projected area of the bearing surface $= 7\frac{1}{2}'' \times 8\frac{1}{4}'' = 61.875$ sq. in. and the pressure per sq. in. of projected area $= \frac{80{,}600}{61.875} = 1{,}302.6$ lbs.

Check for Heating.—From notes on "Bearings" we find the pressure per sq. in. of projected area allowed for cross-head pins to be 750 lbs. and not to exceed 1,000 lbs. This shows that the pins are too small to prevent heating.

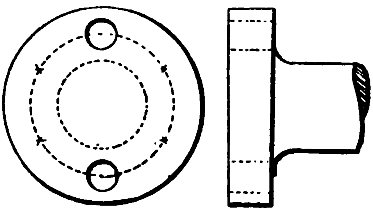
Assume a pressure of 750 lbs. per sq. in. Then the projected area required $= \frac{P}{750} = \frac{80{,}600}{750} = 107.47$ sq. in.

Since $l = 1.1\, d$, $1.1\, d^2 = 107.47$ and $d = 9''.88$ or $9\frac{7}{8}''$. $\therefore l = 1.1\, d = 10''.86$ or $10\frac{7}{8}''$

The projected area is, then, $9\frac{7}{8}'' \times 10\frac{7}{8}'' = 107.39$ sq. in., and the pressure per sq. in. $= \frac{80{,}600}{107.39} = 750.5$ lbs., which is within the limit of 1,000 lbs.

The dimensions of the crank pins are, then:—Dia. $= 9\frac{7}{8}''$; length $= 10\frac{7}{8}''$.

SHAFT COUPLINGS.



The couplings are forged solid with the shafts, and are subjected to a twisting strain tending to shear the coupling off leaving the shaft cylindrical to the ends, so that the coupling can rotate on it like a washer on a bolt. Let t = the thickness of coupling at a radius R from the center of the shaft. Then $2 \pi R t$ is the area of the section and $2 \pi R t f$ its resistance to twisting. Its moment around the center of the shaft is $2 \pi R^2 t f$, and this should equal the twisting moment on the shaft, which is $\pi f d^3 \div 16$. Equating these, we have:— $t = \dfrac{d^3}{32 R^2}$. From this it will be seen that, as R increases, t decreases. When $R = \dfrac{d}{2}$, or where the coupling joins the shaft, we have, $t = \dfrac{d^3}{8 d^2} = \dfrac{d}{8}$. This gives the thickness at this point for strength only. *Seaton* states that, from practical considerations, the thickness of the couplings should not be less than the diameter of the coupling bolts. He also states that, to allow for the decrease in strength due to cutting holes for the bolts, the thickness should not be less than .3 × dia. of a shaft subjected to twisting only.

COUPLING BOLTS.

The number of bolts depends, to a certain extent, on circumstances. If it be desirable to keep the outside diameter of the coupling as small as possible, the number of holes should be increased; this decreases their diameter, and brings their centers nearer to the center of the shaft, thus reducing the outside diameter of the coupling. In general, when engines have two cranks with the shafts in duplicate pieces, there should be an even number of bolts. When there are three cranks and three interchangeable parts of the crank shaft, the number of bolts should be a multiple of three. In this case six is the number usually taken, and this applies to the couplings of the line shaft also.

Diameters of Coupling Bolts.—Let K = radius of bolt circle, which is generally from .75 to .80 of the diameter of the shaft. The coupling bolts, if well fitted, will be subjected to shearing only.

Let f_s = shearing strength of material = $\frac{3}{4}$ to $\frac{4}{5}$ of f_t.

Then T. M. = $\frac{\pi}{16} f_t d^3$, and, at bolt circles, this is $\frac{\pi f_t d^3}{16 K}$.

Let δ = dia. of bolts, and n = No. of bolts, then:—

$$n \frac{\pi}{4} \delta^2 f_s = \frac{\pi f_t d^3}{16 K}. \quad \text{If } f_s = \tfrac{3}{4} f_t,\ \delta = \sqrt{\frac{d^3}{3 n K}}.$$

In the case under consideration, n = 6, K = 10, and d = 13½.
∴ δ = 3½″, about.

I. H. P. REQUIRED FOR GIVEN VESSEL; CYLINDERS; VALVES AND VALVE CHESTS; PISTONS AND RODS; STEAM AND EXHAUST PIPES.

The general method is as follows:—

DATA.—Displacement; length; breadth (extreme); mean draught; area of immersed midship section; speed required.

PROBLEM.—To design the propelling machinery of a naval vessel with the following dimensions and data:—

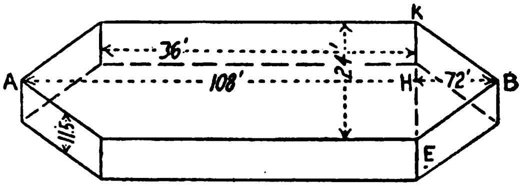
Displacement	850 tons.
Length	180 feet.
Breadth (extreme)	32 feet.
Mean Draught	11.5 feet.
Area of immersed midship section	275 sq. ft.
Speed required	14 knots.

INDICATED HORSE-POWER.—This is determined by two methods:—Kirk's "Analysis," and Froude's "Law of Comparisons."

Kirk's Analysis.—From a comparison of the trial trips of several vessels of similar lines, it is found that, reduced to a general

basis of ten knots per hour, about 7 I. H. P. are required for each 100 sq. ft. of wetted surface.

Wetted Surface.—Kirk's Analysis:



$$850 \times 35 = \text{Vol. in cubic feet.} \quad \frac{850 \times 35}{275} = 108 \text{ ft.} = AH.$$

$$\frac{275}{11.5} = 24' = EK. \quad EB = \sqrt{EH^2 + BH^2}.$$

$$BH = 180' - 108' = 72'.$$

Then, $EB = \sqrt{12^2 + 72^2} = 73'$.

Area of bottom.................	108 × 24 =	2,592	sq. ft.
Area of sides....................	2 × 11.5 × 36 =	828	"
Area of entrance and run........	4 × 11.5 × 73 =	3,358	"
Area of wetted surface...........		6,778	"

For a speed of ten knots 7 I. H. P. are required for 100 sq. ft. of wetted surface, and, as power varies as the cube of the speed, we have:—

$$\textit{Collective I. H. P.} = \left(\frac{14}{10}\right)^3 \times 7 \times \frac{6778}{100} = 1302.$$

From trials of vessels of the same type it is found that the I. H. P. of the auxiliaries is from four to six per cent of the total I. H. P.

Using the smallest value, we have: 1,302 × .04 = 52, and *Indicated H. P. of main engines* = 1,302 — 52 = 1,250.

Indicated H. P. by Froude's Law of Comparisons:—

Froude's Law may be stated as follows: If two ships have the same coefficient of fineness, same water lines, etc., but their dimensions have a ratio *n*, then for speeds s_1, s_2, s_3, etc., of one with resistances r_1, r_2, r_3, etc., the corresponding speeds of the other will be $s_1\sqrt{n}$, $s_2\sqrt{n}$, $s_3\sqrt{n}$, etc., with resistances n^3r_1, n^3r_2, n^3r_3, etc.

In similar ships, the cube of the ratio of dimensions equals the ratio of displacements, hence n^3 = ratio of displacements and $\sqrt{n}$ = (ratio of displacements)$^{\frac{1}{6}}$.

Let d, r, s, and p represent the displacement, resistance, speed, and I. H. P. respectively of a given vessel whose data are known, preferably from a progressive speed trial accurately conducted; or, the same quantities may be obtained experimentally from a model of the proposed ship.

Let D, R, S, and P represent the corresponding quantities for the vessel whose power is to be determined.

$$\text{Then } S = s\sqrt[6]{\frac{D}{d}}, \text{ and } R = r\frac{D}{d}, \text{ also } p = rs.$$

$$\text{Now, } P = RS = rs \times \frac{D}{d} \times \sqrt[6]{\frac{D}{d}} = p \times \frac{D}{d} \times \sqrt[6]{\frac{D}{d}}.$$

Applying this method to the engine under design and assigning as a model the U. S. S. "San Francisco" of similar lines and fineness and whose trial was accurately conducted, we obtain the following:

San Francisco.	*Proposed Ship.*
d = 4,088	D = 850
s = 19.52	S′ = 14
p = 9,718 (main engines).	P′ = ?

Substituting and solving for S and P, we have, S = 15.03 and P = 1,555. This value of I. H. P. refers to the propelling machinery only.

By the conditions given the speed is 14 knots; hence the above may be reduced in proportion.

Indicated Horse-power by Froude's Law $= 1{,}555 \times \left(\frac{14}{15.03}\right)^3 = 1{,}250.$

This is an unusually close approximation to the amount given by the first method.

Indicated Horse-power Used is, then, 1,250.

ENGINES.

The type of engine, number of engines and other data are found from experience and best practice, and for this vessel the following are decided upon:—

Twin screws. Triple-expansion, vertical, direct engines. Independent pumps. Boiler pressure, 160 lbs. Cut-off in H. P. cylinder, .7 stroke. Total ratio of expansion, 7.5. Piston speed per minute, 800′. Stroke, 20″. Revolutions per minute, 240. Vacuum, 24″.

Since there are two engines, the I. H. P. for each will be 625.

DIAMETER OF THE L. P. CYLINDER.

For a ratio of expansion of 7.5, using the tables, *Seaton,* p. 118, we have, $p_m = 175 \times .402 = 70.35$. Back pressure is 3 lbs., so the theoretical mean effective pressure is $70.35 - 3 = 67.35$.

From the data of trial trips, the actual mean pressure is found to be from 50% to 55% of the theoretical. Using 53% as a mean, we have, $p_m = 67.35 \times .53 = 35.7$.

Area L. P. Cylinder $= \dfrac{625 \times 33{,}000}{800 \times 35.7} = 722.16$ and Diameter is 30″.3.

Diameter by Comparison.—Let I. H. P., p_e, p_s, and d represent the I. H. P., effective mean pressure, piston speed, and diameter of cylinder (L. P.) of a given engine, and I. H. P.′, p'_e, p'_s and d′ the same for a required engine.

$$\text{Then } \frac{\text{I. H. P.}}{\text{I. H. P.}'} = \frac{p_e \times p_s \times d^2}{p'_e \times p'_s \times d'^2} \text{ and } d'^2 = \frac{\text{I. H. P.}'}{\text{I. H. P.}} \times \frac{p_e}{p'_e} \times \frac{p_s}{p'_s} \times d^2.$$

If the initial pressure and total expansion are the same in the two engines, then $\dfrac{p_e}{p'_e} = 1$ and $d'^2 = \dfrac{\text{I. H. P.}'}{\text{I. H. P.}} \times \dfrac{p_s}{p'_s} \times d^2$.

Applying the formula and comparing with the U. S. S. "Newark," we have the total ratio of expansion the same and the initial pressure also. Hence, $p_e = p'_e$.

For the U. S. S. "Newark," d = 76, I. H. P. = 4,250, and piston speed = 850.

$$\text{Hence, } d'^2 \; \frac{625 \times 850 \times 76^2}{4{,}250 \times 800}. \quad \text{Then } d' = 30''.04.$$

Again, using the data of the U. S. S. "Yorktown" and class, we have:—

$$\text{I. H. P.} = 1{,}600. \quad d = 50. \quad p'_e = 750.$$

$$d'^2 = \frac{625 \times 750 \times 50^2}{1{,}600 \times 800}. \quad d' = 30''.26.$$

Diameter Taken, L. P. Cylinder.—By the two methods we find that the diameter is between 30″ and 31″, so we take the diameter as 31″.

DIAMETER, H. P. CYLINDER.

$$\text{Area} = \frac{\text{Area L. P. cylinder}}{\text{Cut-off in H. P. cylinder} \times \text{total ratio of expansion}} =$$

$$\frac{31^2 \times .7854}{.70 \times 7.5} = 143.7. \quad \text{Then, } \textit{Diameter} = 13''.5.$$

DIAMETER, I. P. CYLINDER.

For the I. P., we take a mean proportional between the diameters of the H. P. and L. P. cylinders.

Area I. P. cylinder $= \sqrt{\text{area H. P.} \times \text{area L. P.}} = 330$ sq. ins. *Diameter* = 20″.5.

Calculated Diameters.—H. P. = 13″.5; I. P. = 20″.5; L. P. = 30″.25.

Since these calculated diameters are so near the actual diameters of the cylinders, we will change them to those actually used, in order to be able to carry on the design to better advantage for instruction. Then the *Diameters of Cylinder Used* = H. P., 13″.5, I. P., 21″, L. P., 31″.

Ratio of cylinder volumes, $\frac{\text{L. P.}}{\text{H. P.}} = 5.273. \quad \frac{\text{I. P.}}{\text{H. P.}} = 2.42. \quad \frac{\text{L. P.}}{\text{I. P.}} = 2.179.$

Areas of pistons, H. P. = 143″.139. I. P. = 346″.361; L. P. = 754″.7.

THICKNESS OF CYLINDERS.

The H. P. cylinder is calculated for gauge pressure.

Haswell: $t = \frac{P\,D}{2{,}500} + \frac{1}{8}''$. For vertical engines.

P = boiler pressure. D = diameter of cylinder in inches. Substituting, $t = {}''.989$.

In the Bureau of Steam Engineering the thickness is frequently calculated for tension, using a pressure $1\frac{1}{2}$ times the gauge pressure, and a tensile strength of 3,000 for cast iron.

Then, $t = \frac{pd}{2f}$. Substituting, $t = \frac{\frac{3}{2} \times 160 \times 13.5}{6{,}000} = {}''.54$.

To the result obtained a quantity is added varying from $\frac{1}{4}''$ to $\frac{1}{2}''$ (depending upon the size) to allow for reboring. Adding $\frac{3}{8}''$ to the result obtained, we have:— $t = \frac{7}{8}''$.

In Weisbach's "Mechanics of Engineering," the following formula for steam cylinders is given: $t = .00028\ Pd$, obtained by taking the tensile strength of cast iron as 18,000 and allowing a factor of safety of 10. Then: $t = 160 \times 13\frac{1}{2} \times .00028 = {}''.6048$.

Using the second formula, the result obtained, $\frac{7}{8}''$, will give ample strength and stiffness.

Then $\frac{7}{8}''$ is the thickness of the H. P. cylinder.

Thickness of the I. P. Cylinder.—Using the same formula as above, we have:—

$t = \frac{21 \times 60}{2{,}500} + \frac{1}{8}'' = \frac{5}{8}''$, assuming the receiver pressure = 60 lbs.

Thickness of the L. P. Cylinder.—For the L. P. cylinder, owing to the reduced pressure in the 2d receiver (usually only from 10 to 15 lbs. above the atmosphere), the thickness required for strength will be considerably less than the results obtained for the H. P. and I. P. cylinders. Consequently, the principal consideration governing the thickness of the cylinder is stiffness, and an amount should be added to the thickness required for strength sufficient to insure a good casting, and one that will maintain its form. Making these cylinders of the same thickness and equal to that of the H. P., $\frac{7}{8}''$, will insure ample rigidity and strength.

Thickness of all Cylinders.—$\frac{7}{8}''$.

CYLINDER COVERS.

The covers are of cast steel, $f = 10{,}000$.

The use of cast steel for cylinder and valve chest covers was not entirely successful for a time, due principally to the porosity of the castings obtained. At first the thickness was increased as a remedy until the weight was equal to that of the cast-iron covers. Thus, cast steel was only productive of additional expense, and in several instances its use was discontinued for this purpose and cast iron substituted. At present, the makers, having profited by past failures, are able to make these castings sound, and, as the strength of cast steel is considerably greater than that of cast iron, this material is to be used. One of the principal causes of failure was due to blowholes near the flanges, which was remedied by increasing the thickness there.

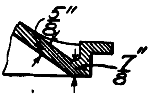
The cover is in the condition of a flat circular plate secured at the edges and uniformly loaded. (See *Unwin,* Part I, p. 112.)

$$t = r\sqrt{\frac{2}{3}\,\frac{p}{f}} = 6.75\sqrt{\frac{2 \times 160}{3 \times 10{,}000}} = {''}.7, \text{ nearly.}$$

The cover must be thickest where it joins the cylinder—that is, at the flange.

Prof. Marks gives as a rule that the flange should be about $\frac{5}{4}$ the thickness of the cover. Then:— $\frac{5}{4} \times {''}.7 = \frac{7}{8}''$.

Sufficient thickness must be given here to insure a sound casting, so this will be increased to 1″, which will also allow for shocks when the water collects in the cylinder.



Thickness of H. P. Cylinder Cover = 1″ at flange and would be $\frac{7}{8}''$ if cover were flat, for the inner part; but, as shown in the sketch, it is found that a thickness of $\frac{5}{8}''$ is amply strong, using strengthening ribs.

Thickness of I. P. Cylinder Cover. Using the same formula,

$$t = 10.5\sqrt{\frac{2 \times 60}{3 \times 10{,}000}} = {''}.6.$$

Thickness of L. P. Cylinder Cover. $t = 15.5\sqrt{\dfrac{2 \times 20}{3 \times 10{,}000}} = {''}.5.$

It is seen that the calculated thickness of the covers for the I. P. and L. P. cylinders is less than is required for sound castings, so that all the covers will be made of equal thickness. By using four strengthening ribs for the H. P. cover and 6 for the I. P. and L. P. covers, we preserve the same strength for the curved covers as we should have had for flat ones, making them ⅞″ thick. The ribs will be made of equal thickness for all covers, ⅜″, and of a depth sufficient to make them flush with the flanges.

Thickness of all Cylinder Covers = ⅝″, and flanges 1″.

BOTTOM HEADS.

These are cast with the cylinder and are cast hollow for the I. P. and L. P., and single for the H. P.

Thickness.—I. P. and L. P., *Seaton,* p. 159: $t = .65\ f$. f is a constant, equal to the thickness of the barrel + ¼″. $t = \frac{7}{8}'' + \frac{1}{4}'' = 1''.125$.

H. P.— $t = 1.1 \times f = 1.1 \times 1''.125 = 1''.237$. Owing to the small diameter and the curved shape, we may make this thickness not greater than ⅝″ by using radial ribs.

For the thickness of the parts of metal around the ports, etc., see *Seaton,* p. 159. It will be noted, however, that the results given are much heavier than those used for naval vessels, so that a considerable reduction may be made throughout.

CYLINDER HEAD BOLTS.

These bolts are designed for a tensile strength equal to or greater than the maximum load on the cover. Then, from the power exerted by a man using an ordinary wrench in setting up the bolts, a certain diameter, not less than about ⅝″ (see *Unwin,* Vol. I, p. 152), is necessary to prevent the bolt from being twisted off.

Also, the cylinder cover bolts must be spaced sufficiently close to make a steam-tight joint. They are seldom spaced more than 5″ apart, and the pitch depends on the size of the cylinder and the pressure to be withstood.

Number of Bolts and Spacing.—Approximate diameter of the bolt circle = diameter of cylinder + 2 × space between bore of cylinder and edge of bolt + dia. of bolts. From the preceding, the bolts must be at least ⅝″ in diameter. Take 1″ as a trial diameter of bolt, and allow ⅝″ between the bore of the cylinder and the edge of the bolts. Then, diameter of bolt circle = 13″.5 + (2 × ⅝″) + 1″ = 15¾″, and the circumference = 49″.88.

An even number of bolts is generally used, and it is seen that 10 bolts would make the space between the bolt centers 4″.98—too near the limit—so that 12 bolts will be used, and the pitch of the bolts will be 49″.48 ÷ 12 = 4″.16.

Having determined by this trial the number of bolts, we now find the diameter needed for strength.

Diameter of Bolts for Strength.—Let x = the diameter of bolts. Consider the steam pressure to act to the bolt circle and use the area of the bolt circle instead of that of the cylinder. Then:—

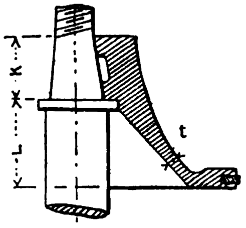
$$12 \times \frac{\pi}{4} x^2 = \frac{(15\frac{3}{4})^2 \times \frac{\pi}{4} \times 160}{f_t = 5{,}000} \text{ and } x = {}''.8.$$

This is, of course, "effective" diameter and for outside diameter, see "Table of Bolts and Nuts," p. I, is 1″.

Number and Diameter of Cylinder Head Bolts.—12 and 1″.

Diameter of H.P. Cylinder Cover.—Allowing ¾″ of metal beyond the bolts, the diameter would be 13″.5 + (2 × ⅝″) + (2 × 1″) + (2 × ¾″) = 18¼″.

HIGH-PRESSURE PISTON.



The pistons are of cast steel and conical in shape. This form gives additional stiffness, and also allows the stuffing box to be placed partially within the lower head, thus securing a longer connecting rod.

The method of proportioning a cast-steel piston is found in *Unwin,* Vol. II. The formula given there for t is from: t = .0015 d $\sqrt{p}$ to t = .003 d $\sqrt{p}$ depending on the size of the piston, but in no case is it to be less than ⅝″.

To find K, refer to *Unwin,* working for the L. P. piston, as the rods are to be interchangeable.

Solving for t in sketch, using larger coefficient, we have: $t = .003 \times 13.5 \sqrt{160} = {}''.51$. As this is less than ⅝″, we make $t = \frac{5}{8}''$.

PISTON RODS.

Diameter: From *Seaton's* general formula, p. 173,

$$d = \frac{D}{55} \sqrt{p\,(1 + ar^2)}$$

D = diameter of the cylinder. p = unbalanced pressure = 175 — 60 = 115 lbs. $a = \frac{1}{3{,}000}$. r = 12 = ratio of length to diameter of rod (assumed).

$$\text{Substituting, } d = \frac{13.5}{55} \times \sqrt{115\left(1 + \frac{144}{3000}\right)} = 2''.67.$$

Second Method.—Seaton's practical formula, p. 174: $d = \frac{D}{F} \sqrt{p}$

For direct-acting naval engines, F = 60. $d = \frac{13.5 \sqrt{115}}{60} = 2''.4$.

Mean of Results, or Diameter of Rod = 2″.53 or 2½″.

By applying these formulæ to the I. P. and L. P. rods, the results obtained will be slightly less. As the rods are to be interchangeable, all the rods will be made of the same diameter, and all other dimensions will agree as well.

Diameter of all Piston Rods.—2½″.

Collars.—The rods are forged with a collar. This collar is to be recessed into the piston, and to be flush with the face of the piston when set up.

Taper of Rods.—Where the rods pass through the hub of the piston they will be given the conventional taper of ¾″ to 1′. This will give a diameter of 2¼″ at the upper end of the taper. The piston is secured to the rod by a nut of standard size for a bolt of 2¼″ outside diameter. The collars for this diameter of rod will be about 3½″ in diameter and about ¾″ in thickness, which will give ample strength to resist shearing and will allow considerable margin to resist shock.

PISTON DETAILS.

Breadth of Ring Face.—Seaton, p. 170: Breadth $= .63 \times x$, where $x = \frac{D}{50}\sqrt{p} + 1 = 3''.7$. This is for a horizontal engine, and does not apply to a vertical one, as such a large face is not necessary. The wear and friction will be greater for a wide than for a narrow face, and a longer cylinder will be needed. About $.25 \times x$ will be a suitable value and will be taken.

Breadth of Ring Face.—$.25 \times 3''.7 = 1''$, nearly.

Thickness of Packing Rings.—This is about one-half the total face, and, as the ring space is filled up with two rings, will give a section $\frac{1}{2}''$ square. About $.15 \times x$ would be an approximate value, which gives $\frac{1}{2}''$.

Piston Face.—Allowing $\frac{1}{2}''$ of metal on each side of the rings, we have $1'' + \frac{1}{2}'' + \frac{1}{2}'' = 2''$.

H. P. CYLINDERS, VALVES, ETC.

Cylinder between Heads.—Length = stroke + clearance, top and bottom, + piston face $= 20'' + \frac{5}{8}'' + 2'' = 22\frac{5}{8}''$.

Shape of Bottom Heads.—This will conform to the shape of the piston. Good metallic packing will be fitted to the rod. The opening in the bottom for packing we will take as 5″. This will vary according to the type of packing used, and the space required from the makers' catalogues. In this case also the diameter of the flange required for securing the stuffing box is taken as $7\frac{1}{2}''$.

Counterbore.—The cylinders should be counterbored at each end, and to a length that will ensure the edge of the packing-ring overrunning at each stroke. The usual depth of counterbore is, for engines of this size, $\frac{3}{16}''$, so that the diameter of the cylinder at the counterbore will be $\frac{3}{8}''$ greater than for the bore of the cylinder. By drawing the piston in position, the point where the clearance should begin is found, the piston to pass over the edge of the counterbore about $\frac{3}{16}''$.

Port Passages.—A line at right angles to the center line of the cylinders at the inner ends of the counterbores will cut the opposite

cylinder walls at the proper points for the inner edges of the port passages where they enter the cylinders.

Port Area for H. P. Cylinder.—Since the steam has to exhaust by the same port that it enters by, the size of the ports should, as a rule, be governed by the velocity of the exhaust steam, rather than by the velocity of the steam at admission. This area should be such that the velocity of the exhaust steam does not exceed 6,000 ft. per minute.

Seaton states that, in compound engines, the exhaust passages from the H. P. cylinder should be such that the velocity of flow is not more than 5,000 ft. per minute, in order that the difference between the pressures of exhaust of H. P. and admission of L. P. may not be excessive.

In general practice, the port area is designed for the exhaust, the valve is designed to open wide for exhaust, and, assuming a velocity of 5,400 ft. per minute, we have:—

$$\text{Area of port} = \frac{\text{Area of H. P. cylinder} \times \text{piston speed}}{5{,}400} = \frac{143.14 \times 800}{5{,}400} = 21.2 \text{ sq. in.}$$

Length and Width of Steam Ports.—The width, parallel to the cylinder bottom, or, at right angles to the axis of the cylinder, is usually taken from .6 to .9 diameter of the cylinder. $.9 \times \text{diameter} = .9 \times 13.5 = 12''.15$. Take this as 12″. Then, the length of the port in the direction of the cylinder axis is $\frac{21''.2}{12} = 1\frac{3}{4}''$.

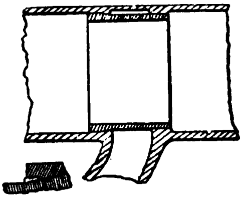
Dimensions of Ports for H. P.—These are then $12'' \times 1\frac{3}{4}''$.

Clearance.—For this size of engine the clearance is taken as $\frac{3}{8}''$.

Diameter of H. P. Piston Valve.—This must be of such a diameter as will give an equal port area.

Measurements of various valves show that about 25 per cent. of opening is an average for the surface taken up by the bridges of the liner. The Bureau of Steam Engineering also requires that no more than this amount shall be used.

The maximum port opening for steam is assumed as 1″, as this amount is found by experience to be about right for this diameter of cylinder and piston speed. The required exhaust opening is found from the valve diagram, which will be discussed later, and in this case, for a perfectly designed valve gear, is $2\frac{1}{8}''$; but it is advisable to reduce this to $1\frac{1}{2}''$ at the expense of increasing the back pressure in the H. P. cylinder. Since, then, $1\frac{1}{2}''$ is assumed as the length of the ports in the liner parallel to the axis of the piston, then: Area of piston valve cross-section × length of opening × per cent. of opening used = area of steam port. Or: $\pi D \times 1\frac{1}{2}'' \times .75 = 21\rlap{.}''2$ and $D = 6''$.



Working Liner H. P. Valve Chest.—There will be a working liner at each end of the valve chest. It will be made of cast steel $\frac{1}{2}''$ thick. This is to be forced into place and held by screws tapped half into the linings at the ends and half into the metal of the chests, and arranged as shown in the sketch. At the ends of the liners, where they are driven in, they are reinforced to a thickness of $\frac{3}{4}''$ and a shoulder is formed.

There is a clearance of $\frac{1}{4}''$ outside of the liner at the ends to allow of the entrance of the liner into the casting.

Diameter of Valve Chest.—Referring to the sketch and adding the dimensions, the greatest internal diameter of the valve chest is $6'' + (2 \times \frac{3}{4}'') + (2 \times \frac{1}{4}'') = 8''$.

Thickness of Chest.—If we suppose the valve chest to be a working cylinder for its whole length and apply the formulæ, the thickness would be about $\frac{5}{8}''$ by Haswell's rule, and by the Bureau method:—

$$\frac{3 \times 160 \times 8}{2 \times 6{,}000} + \text{constant} = \frac{1}{3}'' + \text{constant}.$$

From these results, and considering the stiffness imparted by the liners, and the fact that the valve piston works through but a small distance at each end, $\frac{1}{2}''$ will give ample thickness for strength.

Diameter of Valve Chest Cover.—For reasons given in "Cylinder Head Bolts," we use ⅝″ studs for the cover, though they are over strong as far as calculations go. Allow ⅜″ of metal inside of studs and ½″ outside. Then:—

$$\text{Diameter of cover} := 8 + 2\left(\frac{3''}{8} + \frac{5''}{8} + \frac{1''}{2}\right) = 11''.$$

Number of Studs.—Using 8 studs will give a pitch of about 3½″, which will ensure a steam-tight joint, and the combined sectional area will be more than sufficient for the load on the cover.

Thickness of Valve Chest Cover.—Applying *Unwin's* formula for a circular plate (p. 112, Part I), we have:—

$$t = r\sqrt{\frac{2p}{3f}} = 4\sqrt{\frac{2 \times 160}{3 \times 8{,}000}} = ''.46 \text{ or } \tfrac{1}{2}''. \quad f = 8{,}000 \text{ for cast steel.}$$

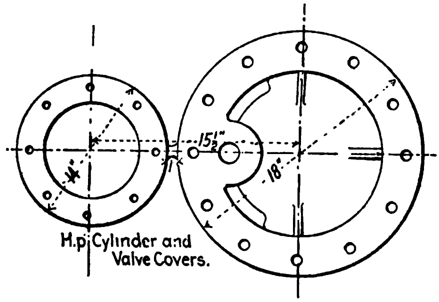
Thickness of Flanges of Valve Chest Cover.—The thickness for the cover must be increased for the flanges for the same reason as in the case of cylinder heads. Then: thickness of cover flanges = ¾″, and for chest flanges ⅞″.

Piston Valves Interchangeable.—All piston valves, covers of chests, and working liners of H. P. and I. P. cylinders will be made interchangeable, using two valves of the same size for the I. P. cylinder.

Crank Shafts Interchangeable.—The crank shaft will be made of three similar parts, united by couplings and bolts.

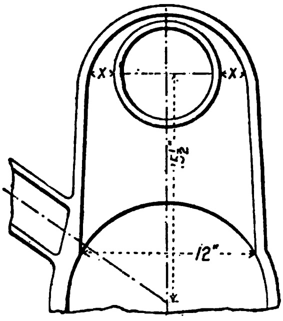
Positions of Eccentrics.—The H. P. and I. P. eccentrics will be keyed to the couplings; that is, the couplings will furnish seatings for the eccentrics instead of the shafts doing so, as is usual. Consequently, the center line of the H. P. valve stem will be determined by the position of the forward coupling.

Center Line of H. P. Valve Stem.—The shafts and bearings have been roughly calculated, and from this the center line of



the forward valve stem is found to be 15½″ from the axis of the cylinder. If the eccentrics could have been bedded close up to the forward bearing, the flanges of the valve chest and cylinder could have been tangent, and this would have given the shortest distance apart of the valve stem and the center of the cylinder, which is always sought for in order to save fore and aft room in the ship. This distance would have been: 9″ + 5½″ = 14½″. Consequently, the flanges, as built, will be separated by a distance of 1″, as in the sketch.

Port to Valve Chest.—To determine the shape and size of the port leading from the valve chest (outside of liner) to the steam cylinder as found previously: The dimensions of the ports at the cylinder walls are 12″ × 1¾″, and its area is 21 sq. in. Referring to the sketch, each quadrant of the liner must furnish one-fourth of the steam; hence, the sectional area at x x must be at least equal to one-fourth of 21 sq. in. or 5¼″. Call y y the length of the circular recess around the liner (see section "Working Liner"). Then x x times y y = 5¼″. For this size of engine and length of valve port, y y may be assumed as 3″. Then x x = 5¼ ÷ 3 = 1¾″.



The total distance across the valve chest through the axis of valve is, then: ½″ + (1¾″ × 2) + 6″ = 10½″.

At the cylinder bore, the port is 12″; hence, joining these points, we have the outline of the passage, as shown in the sketch.

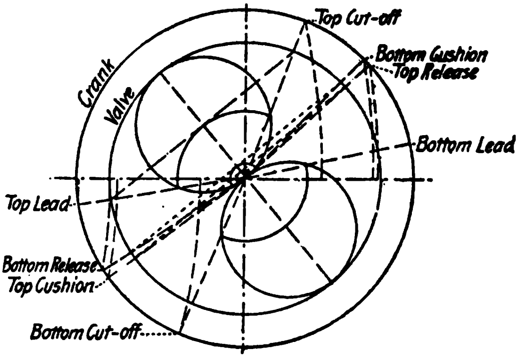
The section will be as shown under "Working Liner," and the port will taper from 3″ at the liner to 1¾″ at the cylinder walls.

Zeuner Valve Diagram for H. P. and I. P. Cylinders:—

Given: Length of connecting rod, 40″. Stroke, 20″.

Assumed: Travel of valve, 4″. Lead angle, top, 9°. Crank position at cut-off, .7 str. In order to allow for the angularity of the connecting rod, the top cut-off is taken at .74 stroke, as the bottom cut-off will be less and the average will be about .7.

Maximum opening for steam, of the port, top, 1″. Release, top and bottom, at $\frac{1}{10}$ of the stroke from the end.



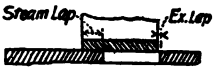
These various assumptions are made from experience.

Scales:

Crank diagram, $\frac{1}{12}$ size. Valve diagram, $\frac{1}{3}$ size.

Found from the diagram: Cut-off, bottom, .65 stroke. Steam lap, top, 1″; bottom, $\frac{29}{32}$″. Exhaust lap, top, — $\frac{1}{8}$″; bottom, $\frac{7}{32}$″.

Steam opening, bottom, $1\frac{1}{32}$″. Exhaust opening, top, $2\frac{1}{8}$″; bottom, $1\frac{25}{32}$″.



Lengths of Working Faces of Valve:

This length must be equal to the opening of the port in the liner + steam lap + exhaust lap. In this case, top = $1\frac{1}{2}$″ + 1″ + (— $\frac{1}{8}$″) = $2\frac{3}{8}$″; bottom = $1\frac{1}{2}$″ + $\frac{29}{32}$″ + $\frac{7}{32}$″ = $2\frac{5}{8}$″.

To Fix the Position of the Inner Edge of the Steam Port.—See next sketch.

As shown later, the diameter of the valve stem, where it passes through the valve, is 1″. From this, the depth of the nut and jam nut for securing the rod to the valve is determined, as shown later, and is found to be $1\frac{5}{8}$″.

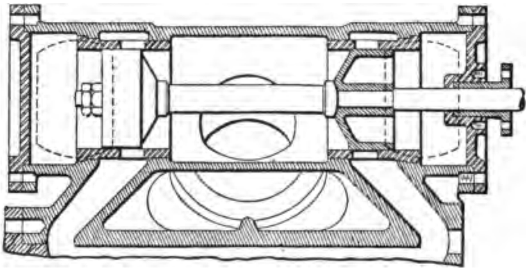
In mid position, the end of the valve stem is distant from the inner edge of the port in the liner, $1\frac{1}{2}$″ (length of port) + 1″ (top steam lap) + $1\frac{5}{8}$″ (thickness of the jam nuts) = $4\frac{1}{8}$″.

Since the travel of the valve is 4″, the travel from the central position in each direction is 2″. Hence, the farthest point that the end of the rod is from the inner edge of the port is $2\frac{1}{2}$″ + $1\frac{5}{8}$″ + 2″ = $6\frac{1}{8}$″.

Adding to this a certain amount for clearance of nut, say $\frac{3}{4}$″, we have $6\frac{7}{8}''$ as the distance from the inner face of the flange of the valve chest bonnet to the inner edge of the port in the liner.

The valve chest cover projects $\frac{3}{8}''$ inside of the flange of the chest; hence, the distance from the inner face of the flange of the chest (which is in line with the flange of the cylinder) to the inner edge of the port in the liner is $6\frac{7}{8}'' + \frac{3}{8}'' = 7\frac{1}{4}''$, which fixes the position of the inner edge of the port.

To allow for alterations, the edge of the port leading to the cylinder should not be flush with the edge of the port in the liner, so they are separated by $\frac{1}{4}''$. Then, the inner edge of the port leading to the cylinder is distant from the inner face of the valve chest bonnet, $7\frac{1}{4}'' + \frac{1}{4}'' = 7\frac{1}{2}''$.

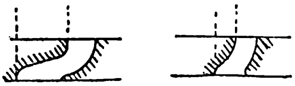


Outer Edge of Flange of Cylinder from the Center Line of the Cylinder Length.—This defines also the inner face of the flange of valve chest bonnet.

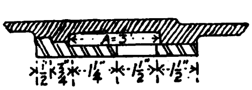
Beyond the counterbore of the cylinder an amount of metal must be left of sufficient depth to allow for securing the $\frac{7}{8}''$ studs of the cylinder heads (as studs must be used at this point). $1\frac{9}{16}''$ is used in this case, so that the flange of the steam cylinder is:—$10\frac{1}{4}''$ (from center of cylinder to edge of counterbore) $+ 1\frac{3}{4}''$ (width of steam port at the cylinder walls) $+ 1\frac{9}{16}''$ (metal for studs) $= 13\frac{9}{16}''$ from the transverse cylinder center line.

Inner Edge of Steam Port at Liner from Transverse Center Line of Cylinder.—This distance is, $13\frac{9}{16}'' - 7\frac{1}{2}'' = 6\frac{1}{16}''$, say $6''$.

From the transverse center line of the cylinder the edges of the ports of the cylinder are $6''$ and $9''$ distant.

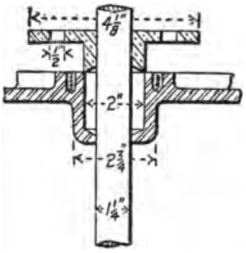


If this distance is made more than 6″—that is, if the ports are shortened—the valve chest will project beyond the ends of the cylinder; if it is made less, the chest will be shortened, but the length of the ports and consequently the clearance volume of the cylinder will be increased.



Length of Liners.—The length should be chosen so that the valve face will overrun at each end, to prevent the formation of shoulders. Length of valve face, 2½″. Travel of valve, 4″. Cutting off ½″ at each end to allow the valve to overrun that amount, the length of the liner = 4″ + 2½″ — 1″ = 5½″. The recess A comes at the middle of the length of the liner and is 3″ long; the remaining dimensions are as in the sketch.

The flanges of the valve chest at the top and bottom ends may be at equal distances from the transverse center line of the cylinder. Hence, the distance 13½″ may be measured either way and the flanges and covers drawn.

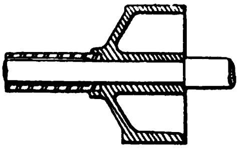


Stuffing-box.—The distance traveled down by the valve from its central position being 2″, the valve stem stuffing-box may be recessed a certain amount from the inner face of the bonnet, this amount depending on the travel + the clearance. Allowing 1″ for the clearance, the stuffing-box may be recessed about 2¼″.

The proportions of gland and stuffing-box may be obtained from *Seaton,* p. 161, using a 1¼″ rod.

Thickness of box	⅜″
Thickness of gland	⅜″
Outside diameter of box	2¾″
Outside diameter of gland	2 ″
Packing space	⅜″
Diameter of flange of gland, elliptical	4⅛″
Thickness of flange	½″

The flange is secured by two ½″ studs, 3″ between centers. These values are taken from the table, p. 161; but, as Seaton states, different manufacturers use different proportions, according to what their practice has shown to be best. The proportions given by any standard writer will usually be satisfactory. With this depth of box the valve will have about 1″ clearance at the bottom.



Valve Laid Off.—From the valve diagram the valve may be laid off in its middle position, the steam and exhaust laps being drawn in as measured from the diagram.

The shape of the valve may be to suit the designer, a flat or conical piston, etc. The thickness of the valve throughout is ⅝″, as this will ensure a sound casting.

Securing Valve Stem.—The valve stem passes through the two pistons of the valve and is secured at the upper end by a securing nut and a jam nut; the diameter is reduced to 1″ where the stem passes through the valve, thus forming a shoulder for the lower part of the valve to abut against. The two pistons are separated by a steel tube distance piece. After drawing the valve in its middle position, the proper length of this distance piece may be found. When the nuts are set up, the distance piece ensures the ends of the valve being the proper distance apart and also allows them to be adjusted on the stem by putting in or removing washers between the distance piece and the valve pistons. The distance piece is recessed into and abuts the two valve pistons, as shown.

With the exception of putting in the steam and exhaust-pipe nozzles, the H. P. valve chest is about finished.

Relief Valves.—At the top and bottom of the cylinder an adjustable 1½″ spring relief valve is placed. For this purpose an opening 1½″ in diameter is made in each head with a suitable boss surrounding it for securing the flange of the valve. The diameter of this boss is 4½″.

Round of Steam Port.—In the cylinder head, if the round of the steam port comes up so as to sensibly diminish the thickness,

an allowance is made by casting a little additional metal on the head over the round of the port, or by curving out the head at that part, as shown in the plan view of the H. P. cylinder.

Lugs for Cylinder Columns.—These will be determined when the three cylinders are designed.

INTERMEDIATE PRESSURE CYLINDER.

Diameter of cylinder, 21″. Area, 346.36 sq. in.

The piston valves, linings, etc., of the H. P. and I. P. are to be interchangeable. Consequently, two piston valves must be used to obtain the necessary port area. With two valves the total port area will be $21.2 \times 2 = 42.4$ sq. in.

Velocity of Steam from I. P. Cylinder.—The velocity of exhaust steam from the I. P. will be $\frac{\text{Area of piston} \times \text{piston speed}}{\text{port area.}} = \frac{346.36 \times 800}{42.40} = 6{,}535$ ft. per min. $= 108.9$ ft. per sec.

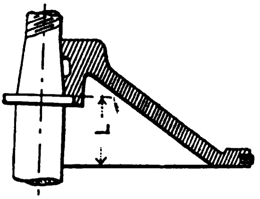
This is not an excessive exhaust-steam speed for this cylinder, so that two piston valves of the same size as used for the H. P. will answer for the I. P. cylinder.

Velocity of Entering Steam.—This will be, as for the H. P., $1\frac{1}{2}$ times that of the exhaust, or $1\frac{1}{2} \times 6{,}535 = 9{,}802$ ft. per min. $= 163.4$ ft. per sec.

Basis of Work.—The facings of the cylinders for the stuffing-boxes will be in line for all the cylinders, giving a basis for laying off the lower heads.

I. P. Piston.—The angle of the I. P. piston is determined from the distance L (which, as the rods are to be interchangeable, must be the same for all pistons) shown in the sketch of the H. P. piston.

In order to make the angle as large as possible, so as to get room for the piston-rod stuffing-box in the lower cylinder head, the part near the boss or hub of the piston —being the reverse of the H. P.—is curved upwards, as shown.



Calculating by Unwin's formula, as in the case of the H. P., the thickness obtained is ¾" near the ring and 1" near the hub.

Dimensions Corresponding to the H. P.:—

Thickness of cylinder flange.
Cavity for the piston-rod nut.
Distance between heads of the cylinders.
Width of cylinder flange.
End of counterbore at each end.
Piston rod and nut.
Depth of piston at center.
Opening for stuffing-box.
Boss and opening for relief valve.

I. P. Cylinder Cover Bolts.—Use, as before, 1" bolts with ¾" of metal outside and ⅝" of metal inside of the bolts.

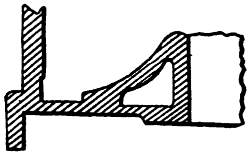
Diameter of flange $= 21'' + (2 \times \frac{5}{8}'') + (2 \times 1'') + (2 \times \frac{3}{4}'') = 25\frac{3}{4}''$.

Diameter of bolt circle $= 21'' + (2 \times \frac{5}{8}'') + (2 \times \frac{1}{2}'') = 23\frac{1}{4}''$.

Circumference of bolt circle = 73.03 inches. Using a pitch a little less than 5" gives 16 bolts, which will give a steam-tight joint, and by calculating the strength, using P = 60 lbs., we find the number ample.

Centers of Piston Valves.—The centers on which the eccentrics are bedded are situated so that the distance between the centers of the I. P. cylinder, and the piston valves is 16½". This is due to the arrangement of the shaft.

By placing the cylinder and valve chest covers about tangent to each other in a plan view and laying off the above distance, the centers of the two piston valves will be 7" above and 7" below the fore and aft center line of the cylinders. (See plan.)

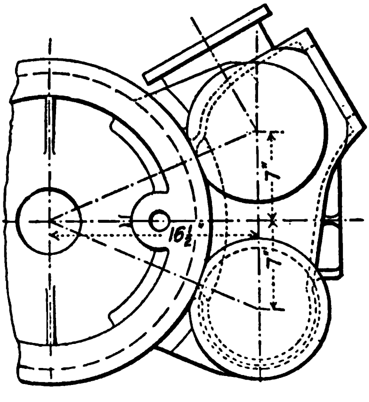


I. P. Bottom Head.—Having designed and drawn the upper end of the cylinder, draw in the bottom head hollow, the upper part conforming to the shape of the lower side of the piston. Make the inner wall of the head ⅝" thick, according to previous notes.

Plan View.—On the center line of the cylinder, projecting from the sectional view, draw the bolt circle and the inner and outer boundaries of the cylinder head flange.

The center line of the piston valves is 16½″ from the center of the cylinder and the center of each valve is 7″ from the longitudinal center line of the cylinder and valve chest. Having fixed these centers, the valve chest covers may be drawn.

To complete the plan, the nozzles for steam and exhaust pipes must be shown and the casings surrounding the valves.



Steam and Exhaust Pipes.—The main steam pipe from the boiler branches to the top and bottom of the H. P. steam chest. The exhaust pipe from the H. P. leaves the steam chest at the middle of its length and branches to the top and bottom of the I. P. steam chest. The exhaust from the I. P. cylinder leaves the steam chest at the middle of its length.

Diameter of Main Steam Pipe.—The velocity of flow through the main steam pipe may be 8,000 ft. per min. Then:—

$$\frac{\text{Area of H. P. cylinder} \times \text{piston speed}}{\text{Velocity of flow of steam}} = \text{Area of pipe.}$$

The cut-off in the H. P. is not considered, as there may be a necessity of following full stroke. Then:

$$\frac{143.14 \times 800}{8{,}000} = 14.31 \text{ sq. in.} = \text{area of pipe.} \quad \text{Diameter} = 4\tfrac{1}{4}''.$$

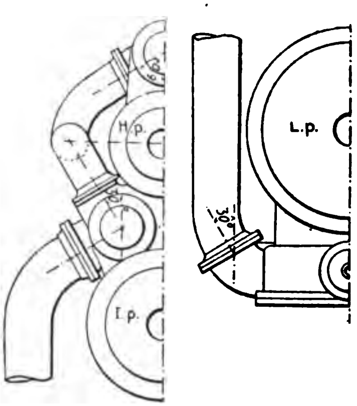
The specifications call for a pipe of 4½″ diameter. As the calculation is near, a diameter of 4½″ will be used, giving less velocity to the steam.

Diameter of H. P. Exhaust Pipe.—The area of this pipe should at least equal the area of the maximum exhaust port opening, of the H. P. cylinder, or 21.2 sq. in. This gives a diameter of 5¼″.

This pipe acts also as a steam pipe for the I. P. cylinder, and the value found for the diameter could be somewhat enlarged with a probable advantage to the engine. Treating the pipe as a steam pipe for the I. P. cylinder, and allowing a velocity of flow of steam of 8,000 ft. per minute, we find that a pipe of diameter of 6½" is required. Its size is to a certain extent regulated by the opening to exhaust of the H. P. cylinder, so that a 6½" pipe is larger than necessary, and, from the two calculations, it is advisable to make the diameter of the pipe part way between, or 5¾" diameter.

Diameter of I. P. Exhaust Pipe.—Area of exhaust from I. P. cylinder = 21.2 × 2 = 42.4 sq. in., and the diameter of the pipe = 7⅜". For the reasons mentioned above, this diameter will be increased to 7¾".

Pipes, H. P. to I. P.—Laying off the plan of the H. P. and I. P. cylinders and valve chest, as in next sketch, and connecting the H. P. and I. P. steam chests by pipes, the shortest and most convenient arrangement will be as shown.



A line at an angle of 60° from the center of the cylinders will give the direction of the H. P. exhaust, and this is curved to meet the steam pipe for the I. P. cylinder beyond the point where it branches to the top and bottom of the chest. For a fair lead this I. P. steam pipe should be at an angle of 30° with the direction of the center line of the cylinders, as shown.

Similarly, the exhaust pipe from the I. P. cylinder has a direction of 60° with the center line so as to clear the cylinder, and then runs parallel with the center line past the L. P. cylinder to the L. P. steam chest, and then curves in at an angle of 30°, as shown, to join the L. P. steam chest.

Angles of Nozzles for Pipes.—The angles made by the flanges for the steam pipe nozzles are now determined, as they are at right angles to the directions of the pipes at those points.

Widths of Flanges for Steam Pipes and Nozzles.—For these consult the standard tables.

I. P. Ports.—Make the width of the ports at the cylinder walls 17¼″, which is a little over .8 the diameter of the cylinder. From previous notes the area of maximum port opening for exhaust is 42″.4; hence, the length of the ports at the cylinder walls should be about $\frac{42.4}{17.25} = 2''.4$. Take this as 2¼″.

Allowing the termination of the counterbore to determine the inner edge of the port of the cylinder, as before, lay off the width of the port. This will be increased to 3″ at the face of the valve chest liner, as in the case of the H. P.

Sections to be Made.—Before making a vertical longitudinal section of the I. P. cylinder and valve chest (section at the valves to be half way between the valves) three other sections will have to be made, as follows:—

A section made by a vertical plane passing through the center of the cylinder and the central line of the valve stem.

A section made by a horizontal plane at the end of the counterbore.

A section made by a horizontal plane passing through the middle of the cylinder and valve chest, and consequently through the center of the exhaust pipe and port.

From the three sections the plan view may be completed, and any portions not hitherto drawn in may be transferred.

Having these three sections, the vertical section on either side of the transverse center line of the cylinder, made by a vertical plane passing midway between the two piston valves, may be drawn.

Securing H. P. to I. P. Cylinders.—The bottom of the I. P. cylinder will be secured to the bottom of the H. P. cylinder by flanges to be drawn in the plan and vertical section and will be 9″ long. The flanges on both H. P. and L. P. will be of such

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width that, when faced off and bolted together, the centers of the cylinders will be 2′ 7½″ apart, due to the arrangements of the shaft.

Bolts.—These cylinder flanges will be bolted together by body bound steel bolts ¾″ in diameter.

Flanges.—The facing of the flange of the H. P. cylinder is 8¼″ from the center line of the H. P. cylinder; hence, the facing of the flange of the I. P. cylinder must be distant from the center of its cylinder 2′ 7½″ — 8¼″ = 23¼″.

LOW PRESSURE CYLINDER.

Having the diameter, length, thickness, etc., this cylinder may be drawn in place in the same manner as laid down for the I. P. The facings of the cylinders for the stuffing-boxes are in line with the others and are of the same dimensions, and the parts mentioned under "I. P. Cylinder" as "Dimensions Corresponding to H. P." have the same dimensions as the H. P. and I. P. At the top of the stroke the face of the piston is laid off from the clearance line, and the angle of the face is determined as before.

Thickness of Piston.—By *Unwin's* rule this is ″.56. For greater security this is increased to ⅞″ near the circumference and 1¼″ near the hub.

Area of Port.—Using a velocity of flow of steam of 8,000 ft.:—

$$\text{Port area} = \frac{784.76 \times 800}{8,000} = 78.46 \text{ sq. in.}$$

.8 × diameter of cylinder = 31″ × .8 = 24″.8. Use 24″ for the width of the port.

Length of Port at the Cylinder. $\dfrac{78.46}{24} = 3''.275.$

A port 24″ wide and 3″ long will give a port area of about 75 sq. in. and a velocity of flow of steam not much over 8,000 ft., and will be used.

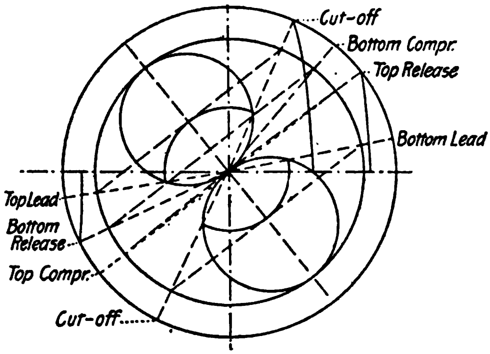
Kind of Valve.—Since the eccentrics are to be interchangeable and are to have an arm of but 2″, a double ported valve must be

used, as the necessary port opening cannot be obtained with a 2″ eccentricity using a single ported valve.

Dimensions of ports at the Valve.—Length of port $= \frac{3''}{2} = 1\frac{1}{2}''$. Width, 24″.

Given: Lead, top, $\frac{1}{4}''$; bottom, $\frac{3}{8}''$. Cut-off, top, .74 stroke. Release, top, .078 stroke from the end; release, bottom, .062 stroke from the end.

Scales.—Crank diagram, $\frac{1}{12}$ size. Valve diagram, $\frac{1}{3}$ size.



Required and Found from the Diagram:

Cut-off, bottom, .64 stroke. Compression, top, .153 stroke. Compression, bottom, .153 stroke. Steam lap, top, 1″; bottom, $\frac{15}{16}''$. Maximum steam opening, top, 1″; bottom, $1\frac{1}{32}''$. Exhaust opening, top, $1\frac{59}{64}''$; bottom, $1\frac{17}{32}''$.

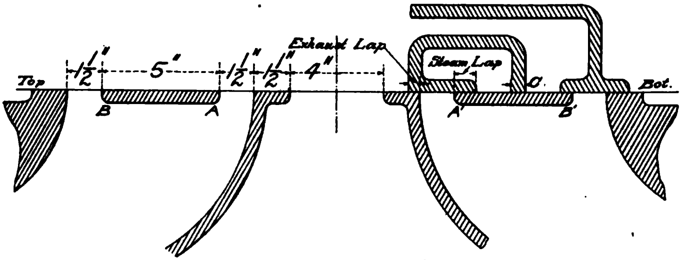
Valve Seat from Central Line of Cylinder.—By trial and making easy curves for the steam ports, the shortest distance of the valve seat from the cylinder central line may be obtained. To facilitate the work, take it, in this case, 21″.

Exhaust Port.—From the transverse center line of the cylinder lay off on each side, on the line for the valve face, the half width of the exhaust port. In this case the exhaust port is 4″ wide, as being amply sufficient for the egress of steam. A general rule for this dimension is that the width of the exhaust port is equal to $\frac{8}{2}$ the width of the steam port. Also, *Unwin* states: "The width of the exhaust port must be equal to or greater than twice the eccentricity, minus the width of the bridge between the steam and exhaust ports."

Cylinder Faces. Thickness of cylinder face over ports. *Seaton* gives two cases—with a false face and without a false face. For the former, $T = 1.2\ F$; for the latter $T = 1.00\ F$, where $F =$ thickness of cylinder $+ \frac{1}{4}''$.

For general use, the latter is preferred for both cases, and it is a matter that is governed by the experience of the designer. Therefore, thickness of cylinder faces $= 1\frac{1}{8}''$.

Length of Bridge between Steam and Exhaust Ports.—This must be long enough to furnish some bearing for the valve to slide on.



Also, care must be taken not to make it so long as to increase the length of the valve chest to an inconvenient degree.

Bridges.—*Unwin,* Part II, p. 225, states: "The length of bridges between steam and exhaust ports is fixed empirically. In small engines it may be $\frac{1}{2}$ the steam port $+ \frac{1}{4}''$, and in large engines it is determined almost entirely with reference to convenience of casting, and should be at least equal to the cylinder in thickness. Take length, $1\frac{1}{2}''$.

Inner Steam Port.—From the edge of each bridge between steam and exhaust, lay off the width of each inner steam port, $1\frac{1}{2}''$, A and A'.

Length of Bridge between Steam Ports.—For this length, A B in sketch, refer to the valve diagram for the L. P. valve. The length of the bridge must be such that, as the valve moves to its farthest position to the right, say, the edge C must not overlap B'. To find this length A B, referring to the sketch, when the

valve is in mid-position, the space it occupies on the bridge A B is equal to:—Steam lap + port in valve + bridge in valve. The farthest movement of the valve to the right, since the valve is to be wide open to exhaust, is:—Steam port + exhaust lap. Therefore, A′ B′ must be equal to this travel, and yet long enough to keep the valve on the bridge, or A′ B′ must be equal to:—Steam port + exhaust lap + steam lap + port in valve + bridge in valve. But, exhaust lap + port in valve = steam port. Hence, A′ B′ = 2 steam port + steam lap + bridge in valve. If the steam lap = bridge in valve, then:—A′ B′ = 2 (steam port + steam lap). Applying to this engine, A′ B′ = 2 × (1½″ + 1″) = 5″.

Balance Piston.—The weight of the valve is taken partly by a balance piston secured to the extension of the valve stem and working in a cylinder open to the condenser at the top. The valve is secured to the stem by a collar and set screw at its lower end and by jam nuts above the balance piston. The valve stem is larger than those for the H. P. and I. P. valves, being 1¾″ in diameter and decreased to 1⅝″ at the valve.

NOTES ON THE DESIGN OF CYLINDRICAL BOILERS.

General Considerations in connection with the design of the ship determine the type of boilers to be used. The type depends upon the vertical distance between the inner bottoms, or reverse frames, and the beams of the deck above; the total length and breadth allowed for boiler space; but principally upon the vertical space.

The decision depends, somewhat, upon a tentative process. If the ship has small draught and is long, the small, low, or gunboat boilers may be used.

If the ship has great beam compared with the draught, the boilers may be single ended, with a single fore and aft fire room, as in the U. S. Ships Miantonomoh and Amphitrite. If the boiler space is sufficient in all respects, the power is divided among three or four furnace, double and single-ended, cylindrical, return-fire, tubular, boilers; or two furnace, cylindrical, single-ended, return-fire, tubular, boilers.

The Conditions Limiting the Diameter are as follows: Sufficient distance must be left between the bottom of the boiler and the inner bottom, or frames, to allow cleaning, painting, and caulking. This is generally six inches for small boilers of eleven feet, or less, in diameter, as the rapid curvature allows these points to be reached. For large boilers this distance is increased to ten and sometimes twelve inches. The distance between sides of boilers and bulkheads is about the same as the distance between shell and inner bottom. This is partly taken up by lagging. The distance between the tops of the boilers and the beams immediately above depends upon the spandrel space required for the uptakes. It should, when possible, be not less than 10″ for boilers up to 11′ in diameter, and as much as 15″ for boilers of larger diameters. The uptakes may extend directly up through the protective deck, in which case no extra space is required for them, or they may join below the deck, as in the U. S. S. "Newark."

The Distance between Adjacent Boilers should be the same, or slightly greater, than that between the boilers and bulkheads.

The Width of Fire Room should be at least one foot greater than the length of the grates to allow the easy handling of fire tools.

Selection of the Type of Boiler, whether high or low, as stated at the beginning of this article, is determined from the class of ship and draught of water, the preference, if there is any choice allowed, being given to the high or return tube type, as less volume of boiler per I. H. P. is required with this type than with the low type.

If the cross-section of the ship will allow of boilers as large as 11′ 6″ in diameter, the high boiler may be adopted; if a diameter of less than 11′ 6″ is all that can be used, the low type must be employed.

This selection of type assumes that the length of the ship for the boiler is not determined, and the type may have to be changed accordingly. If the length of space for one boiler be less than 32′ and more than 21′, the high type must be changed to low, as the double-ended boiler is about 20′ long and the fire rooms at each end must be at least 7′ long. Single-ended high boilers,

placed back to back, take up more length in a ship than one double-ended, high boiler of the same power, as space must be left between the backs, and also, because one single-ended boiler is longer than one-half of a double-ended boiler, owing to the full width between the back of the combustion chamber and shell, and the thickness of the back heads of the boiler in each case.

The low type must be used for lengths between 32′ and 28′, the low boiler being at least 17′ long and the fire room 7′, while the space behind the boiler proper must be at least 4′, to allow for uptakes.

For boiler compartments less than 27′ long, the single-ended high boiler must be used and this type requires a compartment length of at least 19′ as the boiler will be nearly 11′ long, fire room 6′, and the space behind the boiler, 2′.

Grate Area is the first definite quantity found in designing boilers for a ship. It was customary at one time in our service to place in the ship an auxiliary boiler for running the auxiliary machinery while in port. These auxiliary boilers, to use only natural draught, were designed primarily to furnish power for a definite number of auxiliary engines running all the time. Auxiliary machinery is extremely wasteful of steam and runs under conditions of very much less efficiency than the main engines. Some auxiliary engines require from 30 to 40 pounds of steam per I. H. P. per hour, while others require from 160 to 175 pounds. If the amount of steam required by the auxiliary machinery were a constant quantity, it would be easy enough to properly proportion the auxiliary boiler for its work; but, as this cannot be done, the auxiliary boiler has to suffer for the wastefulness of the auxiliary machinery and is now seldom or never fitted in our ships. It is customary, at present, to estimate the grate area necessary to furnish power for all machinery in operation on a trial trip, and to use this grate area as that of the main boilers. Where the calculations for a certain ship allow all boilers to be double-ended, one of these is replaced by two single-ended boilers, both together being of the same power as the double-ended boiler. These single-ended boilers are used in port for auxiliary purposes.

In general, the machinery in operation during a trial trip is as follows:—Main engines; air, circulating, feed, water-service, and flushing pumps; fire room and engine room blowers, and dynamos.

First Method of Calculating Grate Area.—Having determined, from the data obtained on different trial trips, the I. H. P. actually obtained per sq. ft. of grate per hour from boilers similar to those proposed, the grate surface required is determined by dividing the given I. H. P. by the I. H. P. per sq. ft. of grate per hour. The I. H. P. per sq. ft. of grate varies, for one thing, with the air pressure in the fire room, and averages, for an air pressure of 1½″ of water, 16.3 I. H. P. per sq. ft. of grate per hour.

The air pressure allowed by the naval authorities at present is for our cruisers and battleships about one inch of water.

I. H. P. PER SQUARE FOOT OF GRATE ON TRIAL TRIPS.

FOR AIR PRESSURES ABOUT ONE INCH.

	Air Pressure.	*I. H. P. per sq. ft. of Grate.*
Minneapolis	1.000	14.330
Marblehead	1.080	13.160
Massachusetts	1.000	16.900
Oregon	1.000	18.037
Helena	1.000	15.780
Wilmington	1.000	15.030
Iowa	0.987	16.010
Maine	1.074	16.930
Average for 1″		15.680

FOR AIR PRESSURE ABOUT TWO INCHES.

Texas	1.776	16.20
San Francisco	2.020	17.92
Baltimore	2.085	15.82
Newark (1)	2.230	16.42
Newark (2)	2.250	16.90
New York	2.000	16.55
Brooklyn	2.260	18.47
Average	2.090	16.90

Average of both tables above gives 16.3 I. H. P. per sq. ft. of grate with an air pressure of 1½″.

The data above are from official trial trips.

(*Note.*—Experience in the English naval service has shown the desirability of reducing the amount to which their boilers are forced, and their latest specifications for cylindrical boilers provide a total heating surface of not less than 2.5 square feet per I. H. P. at natural draught power, and 12 to 12.5 I. H. P. per square foot of grate, while the forced draught power is limited to 20 per cent. beyond the natural draught power.)

Second Method of Calculating Grate Area.—Having determined from trial trips the number of pounds of coal burned per sq. ft. of grate per hour, and the pounds of coal burned per I. H. P. per hour, the grate surface required is obtained by multiplying the required I. H. P. by the coal required to produce one I. H. P. per hour—this gives the total amount of coal burned—and dividing the product by the number of pounds of coal consumed per hour per sq. ft. of grate.

From 30 to 40 lbs. of bituminous coal can be consumed per sq. ft. of grate per hour with an air pressure of 1″, and 40 to 50 with an air pressure of 2″.

The amount of coal burned depends upon the kind of coal, the amount of pressure of draught, ratio of length of grate to diameter of furnace, ratio of cross-section area of tubes to grate area and height of smoke pipe. The height of smoke pipe has, however, very little effect when forced draught is used.

Reliability of the Two Foregoing Methods.—The first method of determining the grate area is the most accurate. The I. H. P. and the grate area on trials are accurately measured, while the measurement of coal used is not generally made on trial trips, or if it is measured, there is ordinarily not so much accuracy as in the data for the first method.

Another Method for Grate Area, sometimes used as a check upon the first two methods, is as follows:—

$$\text{Grate area} = \frac{\text{lbs. water required per I. H. P.} \times \text{I. H. P.}}{\text{lbs. water evaporated per lb. coal} \times \text{lbs. coal burnt per sq. ft. grate.}}$$

Using 21.6 lbs. water per I. H. P., and 7 lbs. water per lb. of coal gives a fair approximation. From this point on the steps in the process of designing can be best illustrated by a problem.

PROBLEM.

(*U. S. S. Olympia.*)

Required a set of boilers to supply steam of 160 lbs. pressure, to triple-expansion engines, such as are used in modern naval vessels when developing under a maximum forced draught of 2″ of water, 13,500 I. H. P. for four consecutive hours.

The following points are to be observed in the design of these boilers:

(1) Stays must not be placed close enough to prevent tube sheets springing under expansion while steam is being raised.

(2) Stay bolts are to be screwed and the parts between the sheets turned down to the bottom of the thread; the screw stays to be screwed into the sheets and to be further held by nuts at both ends.

(3) The back ends of furnaces to unite to form the lower front plate of the combustion chamber.

(4) All diagonal braces to be fitted with bevelled washers where they intersect the plate and not to be bent.

(5) Tubes to be pitched horizontally and vertically and not zigzag.

(6) Corrugated or some good modern type of strengthened or ribbed furnaces to be used. Least inside diameter 3′ 3″, least outside diameter 3′ 6″, greatest outside diameter 4′.

(7) Diameter of tubes outside 2¼″, ordinary tubes No. 12 B. W. G., stay tubes No. 6 B. W. G.

(8) Ratio of axial length of furnace to rectified length = 1.175.

(9) The front head to be formed of three plates as follows:—

Portion above the tube sheet a segment of a circle, then bent back to a cylindrical surface.

The tube sheet straight across at the top and corresponding to the furnace space at the bottom. The furnace sheet of the front head to be flanged externally to receive the furnace ends and to

conform to the lower part of the front tube sheet. The tube sheet seams of these pieces to be double riveted zigzag.

(10) Between nests of tubes there must be not less than 6″ clear passage, and between the sides of the combustion chamber and shell at least 4½ inches, this to gradually widen to 8″ or 9″ at the top.

(11) Least distance between backs of combustion chambers at bottom about 6″, this to widen to about 10″ at the top.

(12) Manholes to be placed in each spandrel at the bottom, over the middle furnaces, and in upper head sheet. Principal manholes to be not less than 12″ × 15″.

(13) The top row of tubes to be on a line that divides the circular section into ⅓ and ⅔, or a little above this if necessary to provide the proper heating surface.

(14) A trial sketch to be made to determine the number and arrangements of tubes.

(15) Two furnaces at each end to be connected to a common combustion chamber.

Note.—Some of the foregoing requirements must be met by any marine boiler, others such as (6), (8), (9), and (15) refer to the particular boiler in hand.

CALCULATIONS FOR U. S. S. OLYMPIA.

Grate Area—First Method.—Assuming 16.3 I. H. P. per sq. ft. grate.

$$\text{Grate area} = \frac{13{,}500}{16.3} = 828.2 \text{ sq. ft.} \quad . \; . \; . \; . \; . \; . \; . \quad (1)$$

Grate Area—Second Method.—Assuming 3 lbs. of coal per I.H.P. on forced draught trials and 50 lbs. per sq. ft. of grate, we have:—

$$\text{Grate area} = \frac{13{,}500 \times 3}{50} = 810 \text{ sq. ft.} \quad . \; . \; . \; . \; . \; . \quad (2)$$

Third Method.—The Yorktown's water consumption as accounted for by the indicator was 17.5 lbs., per I. H. P. per hour. The actual water supplied was not ascertained, but supposing a loss from condensation, leakage, etc., of about 20 per cent., the water used per I. H. P. $= 17.5 \div 80 = 21.8$. Taking 50 lbs. coal and assuming 7 lbs. of water evaporated per lb. coal, we have

$$\text{Grate area} = \frac{13{,}500 \times 21.8}{50 \times 7} = 848 \text{ sq. ft.}$$

As the first method is the most reliable, and in this case happens to be about a mean between the last two, we will decide upon 828.2 sq. ft. as the necessary amount of grate surface.

Size and Number of Furnaces.—Before it is possible to fix upon the number of furnaces, the length of grate must be decided, for trial at least, and also the diameter of the furnace. Furnaces of cylindrical boilers should not be less than 30″ in diameter, nor more than 48″ unless in exceptional cases. Wherever possible, a diameter of not less than 40″ should be given as, with smaller furnaces (owing to thickness of fire being practically constant for all diameters) the space above the fuel is much contracted and combustion less perfect in consequence. Long grates cannot be properly worked by hand and are not so efficient in consequence. The length of grate varies from 5′ to 7′ and depends much upon the diameter of the furnace. In our service the average is between 6′ and 6′ 6″.

The specifications for this ship requires a furnace of 39″ least inside diameter. Assuming a length of grate of 6½′ we have:—

$$\text{No. furnaces} = \frac{828}{3\tfrac{1}{4} \times 6\tfrac{1}{2}} = 39 +;$$ taking 40 furnaces we find

$$\text{the actual length of grate} = \frac{828}{40 \times 3\tfrac{1}{4}} = 6.'37 = 6'\ 4''.$$

We have, therefore, for the required boilers, 40 furnaces, 6′ 4″ × 3′ 3″.

Note.—The plans of the boiler space for this ship show that a boiler greater than 15½′ cannot be used; boilers over 15′ in diameter have four furnaces if single-ended and eight if double-ended. We can then have ten single-ended boilers or five double-ended boilers, provided the boiler space will accommodate this number of double-ended boilers. A reference to the plans of the

boiler space shows that four double-ended and two single-ended boilers can be placed. This simply amounts to calculating for five double-ended boilers and cutting one of them in two. In any case the total number of furnaces should be a multiple of twice the number in one head.

Boilers up to 9′ diameter may have 1 furnace.
" " 13′ 6″ " " 2 furnaces.
" " 15′ " " 3 "
" beyond 15′ " " 4 "
in each head.

Approximating the Length of One Boiler.—Allow 12″ in length of furnace for the bridge-wall, and allow for furnace door, flanging, etc., a distance of 4½″ from the outside of front tube sheet to the commencement of grate. This distance is simply taken in order to get a basis of work from some fixed point of the boiler. In this case the outside of the front tube sheet is taken. The furnace ends will still project beyond this a sufficient amount to allow for riveting the furnace tubes to the head of the boiler.

Length of Combustion Chambers.—When two furnaces empty into the same chamber, this is usually about ⅓ of the length of the grate.

For this dimension, practice seems to vary within considerable limits. (See *Seaton,* p. 460.)

Recent practice in the Bureau of Steam Engineering is to make the length of combustion chamber, for furnaces of 40″ in diameter, two furnaces emptying into the same chamber, about 30″, but in any case space enough for a man to work in must be allowed. Make water back between opposite combustion chambers 6″ at bottom, increasing to 10″ at top.

Combustion chamber sheets for boilers working with over 70 lbs. pressure are made of ½″ plates.

Then, approximately, the total length of the double-ended boiler is 2 [4½ + furnace + Combust. chamber + comb. cham. sheet] + 6″ = 2 [4½ + 7′ 4″ + 2′ 6″ + ½″] + 6″ = 21 feet.

Length of Tubes, from the above, outside of tube sheets being distant from each other 7′ 4″ + 4½″, is 7′ 8½″.

Approximate Diameter of Boiler.—*Seaton,* p. 393, says good results (for forced draught) can be obtained with an allowance of $1\frac{1}{4}$ to $1\frac{1}{2}$ cu. ft. of boiler volume per I. H. P.

$40 \div 8 = 5 =$ No. boilers.

$\frac{13,500}{5} \times 1\frac{1}{2} = 4,050$, volume of one boiler in cu. ft.

$\frac{13,500}{5} \times 1\frac{1}{4} = 3,375$, volume of one boiler in cu. ft.

$\frac{4,050}{21} = 192.8 =$ area of head in sq. ft.

$\frac{3,375}{21} = 160.7 =$ area of head in sq. ft.

For an area of 192.6, diameter = 15′ 6″.

For an area of 160.7, diameter = 14′ 3″.

So that a diameter of 15′ 3″ just inside the limiting diameter will give good results: ∴ we make the diameter = 15′ 3″.

The *Number of Furnaces* in one head, by reference to *Seaton's* tables, or by plotting the circle of the head, and roughly putting in the furnaces, is found to be four.

Thickness of Shell:—

$$\frac{t}{r} = \frac{p}{f}$$

factor of safety = 4.5.
p = 160; f = 58,000 to 62,000 for mild steel;
percentage strength of joint = 85.

$$\text{Thickness, } t = \frac{160 \times 183 \times 4.5}{62,000 \times 2 \times .85} = 1''.25.$$

Board of Supervising Inspector's rule:—

$$.80\,P = \frac{f\,t}{6 \times \text{rad.}}$$

$$t = \frac{.8 \times 160 \times 6 \times 183}{62,000 \times 2} + \frac{1}{16} \text{ (for corrosion).}$$

$$= 114 + \frac{1}{16} = 1.21.$$

P = pressure in lbs. per sq. in.
t = thickness in inches.
f = strength of material.

∴ Thickness of sheets = $1\frac{1}{4}''$.

Area of Grate: For One Furnace = 6′ 4″ × 3′ 3″ = 20.58 sq. ft.
20.58 × 8 = 164.6 sq. ft. for one boiler.

Heating Surface of Furnace.—The ratio of length of corrugation to length of flue will be about 1.175.

$$\therefore \text{Heating surface} = \frac{8}{2} \times \frac{41 \times 3.1416}{12} \times 7\tfrac{1}{3} \times 1.175 = 370$$

sq. ft.

Heating Surface of Combustion Chamber.—By measuring a rough sketch by means of a planimeter, it is found that the heating surface of the combustion chambers at one end is slightly in excess of the heating surface of the four furnaces emptying into them. This surface for this boiler is estimated at about 400 square feet.

The Total Heating Surface that a cylindrical boiler may contain is = [diameter of shell]^2 × [length of tube + 3] for a single-ended boiler. For a double-ended boiler this will be changed to 2 × [length of tube + 3]. From the above rule of *Seaton's*:—

Total heating surface = $(15\tfrac{1}{4})^2 \times 2 \times (7\tfrac{17}{24} + 3)$ = 4,981 sq. ft.

In modern practice the heating surface varies from 2 to 2.3 sq. ft. per I. H. P. While considerable care and skill are required to keep it as low as 2 ft. and still obtain good results, 2.3 is of common occurrence. Taking a mean we have 2.15 sq. ft. per I. H. P., and for one boiler, 2,700 × 2.15 = 5,805 sq. ft.

Tube Heating Surface = 5,805 — 370 — 400 = 5,135 sq. ft.

The *Area through Tubes* should be about $\frac{1}{7}$ the grate area, with natural draught. *Seaton,* p. 392. In modern boilers designed by the Bureau of Steam Engineering the calorimeter through tubes is nearer $\frac{1}{8}$. For a mean, take ratio of grate surface to area through tubes of 7.5.

Tube area = 164.6 ÷ 7.5 = 21.94.

Diam. tubes, outside = 2¼″ thickness 12 B. W. G. about ″.12.

Internal diameter = about 2″.

$$\text{Area through one tube} = \frac{\pi d^2}{4 \times 144} = \frac{3.1416 \times 4}{4 \times 144}.$$

$$\textit{Total Number of Tubes} = \frac{164.6}{7.5} \times \frac{144}{3.1416} = 1{,}008.$$

Number of tubes in each end = 504.

Pitch of Tubes, about 1.5 d and greater if possible. 1.5 d = 3⅜″ = 3½″.

This distance 3½″ is the standard practice of the Bureau of Steam Engineering for a 2¼″ tube. Pitching these tubes on the tube sheet, about 532 can be placed, as shown, in the four nests. Allowing ample space between nests and over furnaces, number of tubes actually put in each end of a boiler = 532.

In calculating the tube heating surface the total length of the tube was taken and the portion of the tube sheet between the tubes was neglected in calculating the combustion chamber heating surface.

Length of tube = 7′ 8½″, circumference of 2¼″ = 7″.0686.

$$\text{Tube H. S.} = \frac{2 \times 532 \times 92.5 \times 7.0686}{12 \times 12} = 4{,}831.$$

$$\text{Total H. S.} = 4{,}831 + 370 + 400 = 5{,}601 \text{ sq. ft.}$$

$$\text{Ratio of H. S. to grate surface} = \frac{5601}{164.6} = 33.82.$$

Laying Down Boiler on Drawing Board.—Draw a transverse view of shell and divide it by a vertical center line. One-half of this view is to be shown in section (the section taken between front head and front combustion chamber sheet), the other half in elevation. From the center, describe the arc of centers of furnaces. Leave 2″ between the outer surfaces of furnaces. This is practically settled by the fact that the front head sheet must be flanged outward to receive the furnaces and a certain distance is required for this operation. The furnaces are 45″ from center to center, measured on chords of this circumference. The two middle furnaces have their centers 22½″ from the vertical line. Since the corrugations of the furnace alternate, one furnace in the sectional view will be shown through the bottom of the corrugations 39″ in diameter, and the adjacent furnace will show the maximum diameter 43″. Draw the sectional view of the combustion chamber with a water space of 4″ between it and the shell at the bottom, increasing to 7″ near the top of the tubes. Between the two combustion chambers allow a 6″ water leg.

Between each nest of tubes over the furnaces allow about 6″ clear for circulation, examination and cleaning. As this boiler will require three sheets to make up one circumferential belt, we must locate the longitudinal joints. In order to be clear of obstructions, let one joint be about 45° with the vertical, and the remaining joints 120° from this one; the joints of successive belts breaking joints with one another.

The longitudinal joints will be double strapped butt joints and the details will be found under Rivets and Riveting.

The tubes will be of mild steel $2\frac{1}{4}''$ external diameter, No. 12 B. W. G. and swelled to $2\frac{5}{16}''$ at front end. The back end of tubes will be expanded into the tube sheet, beaded over into a counterbore, which will be filled with a ring for protecting the ends of the tubes from the action of the flame. Any mode of protection may be used, subject to the following conditions:—

1. That it will not interfere with the use of ferrules.

2. Will not cause injury to the tube sheets when the tubes are cut out.

3. Will not reduce the area through tubes.

Stay Tubes.—These will be No. 6 B. W. G. thick and will be reinforced at both ends to an external diameter of $2\frac{3}{8}''$, the bore of the tubes being uniform from end to end. They will then be swelled out at the front end to $2\frac{1}{2}''$. The stay tubes are threaded parallel at the combustion chamber end and taper at the front end to fit threads in the tube sheets. They will be screwed into the front tube sheet to a tight joint and will be made tight at the back end by expanding and beading.

The holes for stay tubes are tapped at one operation by a pair of taps on a rod as shown in Fig. 1, and a duplicate of this is always carried among the spare machinery of the ship.

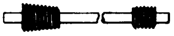


FIG. 1.

Cast-iron ferrules will be used to protect the ends of the stay tubes at the combustion chamber ends. The space between tubes should be sufficient to permit free circulation of water and allow the steam to rise freely to the surface. The tubes should be arranged in horizontal or nearly horizontal rows, *i. e.,* they should not be zigzag. They should not be crowded, as the steaming capacity of the boiler may be seriously impaired thereby.

The top of the top row of tubes should not be placed nearer to the top of the boiler than at a distance equal to one-third of the internal diameter of the boiler.

$$\frac{15'\,3'' - 2''.5}{3} = 5' \text{ to top row of tubes.}$$

From this the center line of the top row of tubes can be obtained and the tubes pitched horizontally and vertically. In laying out the tubes, begin at the lower row, leaving 6″ clear above the furnace for wing furnaces and 13″ for the bottom furnaces. This is done for practical considerations and to give free outlet for the bubbles of steam forward on the crown sheet. The top and inboard side of the combustion chamber can then be completed.

Having put in the furnaces, combustion chambers, and shell, in the transverse view, lay off the divisions into which the head sheet is divided. The upper sheet is cut with an allowance for lap sufficient for a double riveted lap joint (zigzag) leaving just sufficient metal clear above the tubes to permit of expanding the tubes (½″ to 1″ will be sufficient). Lay out the clear dimensions of the tube sheet for similar riveting at top and bottom.

In cutting the lower part of the front tube sheet let it enclose the manholes to be placed in the wing spandrels and above the middle furnaces, and thus act as a strengthening ring. The lap around these manholes should be sufficient for single riveting. In cutting the tube sheet from the wing to the middle nest of tubes, let the slope be about 30°.

The manholes in the wing spandrels will be three-sided, about 13″ × 15″, with the corners rounded and the sides curved to conform to the curves of the shell and furnace. The center manholes are to be elliptical, 11″ × 15″. Between the center furnaces and nests of tubes, support the flat head by braces about 10″ apart, three over each furnace and one at the corner of the wing manholes. These braces tie the head sheet to the front plates of the combustion chambers. In the lower spandrels cut a 11″ × 15″ manhole; and at each side of these manholes and at the top, strengthen the head by a brace; the upper one to continue through

to the opposite end of boiler or to the combustion chamber; the lower ones to be diagonal and connected to the shell.

The back tube plates vary in thickness from $1\frac{9}{16}''$ to $\frac{5}{8}''$, *Seaton*, p. 460. They are generally made, however, from $\frac{5}{8}''$ to $\frac{3}{4}''$, depending on the size of boiler, as with this thickness the tubes can be made quite tight, and there is less liability of cracking the plates or burning the tube ends than with thicker plates.

It is not usual to make the sides and bottom of the combustion chamber exceed $\frac{1}{2}''$ in thickness. They are stayed by screwed stays and the bottoms are stiffened by angle irons, about $3\frac{1}{2}'' \times 3\frac{1}{2}'' \times \frac{1}{2}''$, riveted to them. The screw stays are pitched to suit this thickness of plate by the method given in *Unwin*, p. 112.

The increased pressures have led to the adoption of corrugated, ribbed and spiral furnace flues for additional strength against collapse without abnormally increasing the thickness of the plate. There is a great objection among engineers to furnaces over $\frac{1}{2}''$ or $\frac{5}{8}''$ thick, on account of the increased resistance to the transmission of heat through the plate and the consequent overheating. While the resistance to the transmission in a laminated material may be abnormally increased by the thickness, yet in a homogeneous metal like steel it would seem that the thickness ordinarily used might with safety be increased, for, from experiment, the *condition of the surfaces* seemed to have a much greater value than the internal conductivity.

From the measured rate of transmission along bars and plates, when fairly clean and free from laminations, it was found that a furnace plate might be increased from $\frac{1}{2}''$ to $\frac{3}{4}''$ without increasing the resistance more than 1.25 per cent.

In general, the practical thickness used for 160 lbs. pressure is $\frac{1}{2}''$ and $\frac{5}{8}''$.

The following is the formula generally adopted for corrugated flues:—

$$p = \frac{C \times t}{d}$$

t = thickness of plate in inches.
p = working pressure in lbs. per sq. in.
d = mean diameter of furnaces in inches.
C = constant having values from 12,500 to 14,000.

This value of C will depend upon the kind of furnace and of course upon the strength of the metal used. These furnaces are always made of steel of about 27 tons ultimate strength. For an intelligent assignment of a value to C, it is safest to obtain the latest edition of the maker's circular or to consult the value of C as assigned by authorized rules of inspection.

In the case of the boilers in hand,

$$t = \frac{160 \times 41}{12{,}500} = \tfrac{9}{16}''.$$

Riveted Joints.—In general, for thin plates, where two plates of unequal thickness come together, it is customary to proportion the joints to the thinner plate. This rule will not always answer for thick plates, and to secure tightness at the joint it is proportioned, if not for the thinner plate, for a mean proportional between the two.

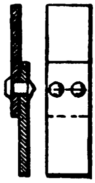


Fig. 2.

For the method of obtaining the formula used in proportioning joints see notes on riveting, p. 118. They are mainly from Trail and Foley and are given as representing the best modern practice and are applicable to thick and thin plates.

For the *Connection of Back End of Furnaces to Combustion Chamber,* and for the joints of adjacent ring furnaces, use a single riveted lap joint. See Fig. 2.

$$t = \tfrac{1}{2}''; \ \% = 54.73.$$

$$d = \frac{1.55 \times 54.73 \times \frac{1}{2}}{45.27 \times 1 \times 1} = {''}937 = \tfrac{15}{16}''.$$

$$\frac{100\,(p - d)}{p} = 54.73. \quad \therefore\ 45.27\,p = 100 \times \tfrac{15}{16}''.$$

$$\therefore\ p = 2\tfrac{1}{8}''.$$

$$\text{Lap sheet} = 2\,E = 2 \times 1.5\,d = 3\,d = 3 \times \tfrac{15}{16} = 2\tfrac{13}{16}'.$$

For the Upper Head Sheet and Tube Sheet.—Tube sheet $= \frac{7}{8}''$, upper head sheet $= 1\frac{1}{4}''$. Making rivet section and plate of equal strength, $\% = \%_1$, and $d = 1''.25$ = thickness of sheet.

The joint is double riveted, lap (zigzag). See Fig. 3.

$$p = \frac{23 \times A \times n \times c}{28 \times t} + d = 2.308 + 1.25 = 3.558 = 3\tfrac{9}{16}''.$$

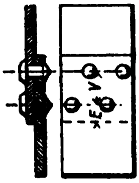


Fig. 3.

In the above, t is taken as $\frac{7}{8}$, that is, the rivet is proportioned to the thinner sheet.

$$\text{Strength of joint} = \frac{3\tfrac{9}{16} - 1\tfrac{1}{4}}{3\tfrac{9}{16}} = 64.9\%.$$

Rivet section, $\%_1 = 64.7$.

$E = \frac{3}{2} \times 1\frac{1}{4} = 1\frac{7}{8}''$.

$$V = \frac{\sqrt{(11p + 4d)(p + 4d)}}{10} = \frac{\sqrt{(39\tfrac{3}{16} + 5)(3\tfrac{9}{16} + 5)}}{10} = 1''945.$$

$V = \frac{15}{16}''$. Lap of sheets $= 2\,E + V = 5\frac{11}{16}''$.

Connection of Lower Edge of Tube Sheet to Head Sheet: double riveted, zigzag, lap, joint.

$d = t = \frac{7}{8}''$.

$$p = \frac{23 \times A \times n \times c}{28 \times t} + d = 1.129 + .875 = 2''004.$$

The percentage strength of plate and rivet sections are:—

$\% = 56.56$, plate and $\%_1 = 56.44$, rivet.

If the lower head sheet had been taken at $\frac{3}{4}''$ and the joint made for equal strength of plate and rivet sections, the joint would more nearly have approached that of the upper one. For example, let $t = \frac{3}{4}''$ (thickness of lower head),

$$\therefore p = \frac{23 \times .60132 \times c \times n}{28 \times .75} + .875 = 2\tfrac{1}{4}'' \text{ nearly,}$$

and the strength of plate $= \left(\frac{2\frac{1}{4} - \frac{7}{8}}{2\frac{1}{4}}\right) 100 = 61.1\%$, while

the strength of rivets $= \dfrac{100 \times 23 \times 2 \times .60132}{28 \times 2\frac{1}{4} \times \frac{3}{4}} = 60\%$.

This latter proportion for the joint would probably be better, hence the lower plate could be made $\frac{3}{4}''$ and a very fair joint made. This sheet being thinner would also make the flanging for the furnaces easier.

Longitudinal Shell Joints: Butt joints, double strapped, treble riveted with alternative rivets omitted in the outer row. See Fig. 4.

Percentage strength of joint = 85.

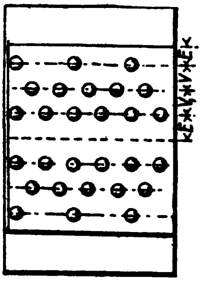


Fig. 4.

$$d = \frac{1.55 \times .85 \times t}{15 \times 5 \times 1.75}; \quad t = 1\tfrac{1}{4}''.$$

$$\text{pitch, } p = \frac{1.55 \times .85 \times 1.25 \times 100}{15 \times 15 \times 5 \times 1.75} = 8''.36,$$

$$\text{or } \frac{p - d}{p} = .85.$$

$$.15\,p = 1\tfrac{1}{4}. \quad \therefore\ p = \tfrac{5}{4} \times \tfrac{100}{15} = 8''.33.$$

$p = 8\tfrac{3}{8}''$, greatest pitch or pitch of outer row.

Middle Circumferential Joint: Treble riveted, lap, zigzag.

$t = 1\tfrac{1}{2}''$, $d = 1\tfrac{3}{8}''$. See Fig. 5.

$$p = \frac{23 \times 1.484 \times 3 \times 1}{28 \times 1.25} + 1\tfrac{3}{8} = 2.93 + 1.375 = 4.305 = 4\tfrac{5}{16}''.$$

$$V = \frac{\sqrt{(47\tfrac{7}{16} + 5\tfrac{1}{2})\,(4\tfrac{5}{16} + 5\tfrac{1}{2})}}{10} = 2.28 = 2\tfrac{5}{16}''.$$

$E = \tfrac{3}{2}\,d = 2\tfrac{1}{16}''$.

Lap of sheet $= 2\tfrac{5}{16} + 2\tfrac{5}{16} + 2\tfrac{1}{16} + 2\tfrac{1}{16} = 8\tfrac{3}{4}''$.

Strength of above joint.

$$\% = \left(\frac{4\tfrac{5}{16} - 1\tfrac{3}{8}}{4\tfrac{5}{16}}\right) 100 = 68.1, \text{ plate.}$$

$$\%_1 = \frac{100 \times 23 \times 3 \times 1.4849 \times 1}{28 \times 4\tfrac{5}{16} \times 1\tfrac{1}{4}} = 67.9, \text{ rivet.}$$

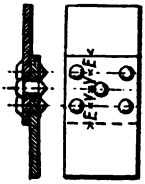


Fig. 5.

Circumferential Seams at Boiler Ends.—Owing to the additional stiffness imparted by the heads of the boiler, these seams are double riveted instead of treble riveted as in the middle seams. The joint is double riveted, zigzag. In this case as the shell is $1\tfrac{1}{4}''$ and the tube sheet $\tfrac{7}{8}''$ the conditions are the same as for the joint between the tube plate and the head sheet, and the same proportions will be used. $p = 3\tfrac{9}{16}''$; $d = 1\tfrac{1}{4}''$; lap $= 5\tfrac{11}{16}''$; $V = 1\tfrac{15}{16}''$.

Width of Butt Straps, Distance between Rows of Rivets, etc. From center of rivet to edge of sheet $= \frac{3}{2}\,d = 1\frac{7}{8}''$.

$$V = \frac{\sqrt{(11\,p + 4\,d)\,(p + 4\,d)}}{10},$$

in which p is ½ greatest pitch $= 4\frac{3}{16}''$.

$$V = \frac{\sqrt{(46\frac{1}{16} + 5)\,(9\frac{3}{16})}}{10} = 2.168 = 2\frac{3}{16}''.$$

$$V_1 = \frac{\sqrt{(11\,p + 20\,d)\,(p + 20\,d)}}{20},$$ where p is the pitch of the outer row or $8\frac{3}{8}''$.

$$V_1 = \frac{\sqrt{(92\frac{1}{8} + 25)\,(8\frac{3}{8} + 25)}}{20} = 3.16 = 3\frac{3}{16}''.$$

Width of butt strap $= 4\,E + 2\,V + 2\,V_1 = 7\frac{1}{2} + 4\frac{3}{8} + 6\frac{3}{8} = 18\frac{1}{4}''$.

$$\text{Plate section} = \frac{8.375 - 1.25}{8.375} = 85.37\%.$$

$$\text{Rivet section} = \frac{23 \times 1.227 \times 5 \times 1.75 \times 100}{28 \times 1.25 \times 8.375} = 84.26\%.$$

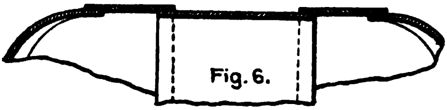
Thickness of Butt Strap.—

$$T = t \times \frac{5}{8}\left(\frac{p - d}{p - 2d}\right) = .95 \text{ or } 1'' \text{ for each strap.}$$

Longitudinal Section of Boiler.—The length of the boiler is, from the details as fixed in the preceding work, 20′ 10″ and is made up of three rings. The end seams are double riveted and the middle seams treble riveted. The distance from the outside of the tube sheet to the edge of the shell sheet is 4″. The lap of the middle seams, previously calculated, is 8¾″. Hence, the length of one ring:—

$$= \frac{20'\,10'' + 2 \times 8\frac{3}{4} - 2 \times 4''}{3} = \frac{21'\,7\frac{1}{2}''}{3} = 7'\,2\frac{1}{2}''.$$

Draw the longitudinal view of the shell, cutting the sheets as shown in Fig. 6.



The upper ends are to be rounded to avoid extensive bracing. Leave a sufficient amount of flat space immediately above the upper joint of the head sheet for one row of braces with their washers, and then round the sheet to an arc of a circle (quadrant): a radius of about 3′ 7″ will effect this.

NOTE.—Curved ends of Cylindrical Boilers—

R = Radius of curved end in inches.

T = thickness of tube plate in 16ths of an inch.

d = outside diameter of tubes.

p = pitch of tubes.

r = ratio of solid metal left between tubes to pitch of tubes, *i. e.*,

$$r = \frac{p - d}{p}.$$

P = boiler pressure.

$$\text{Then } R = \frac{C \times r \times T}{P}, \text{ where } C = 13{,}000.$$

If the percentage strength of the horizontal seam above the tubes be less than the percentage strength of plate left between the tubes, such lesser percentage should be used in calculating R. If the plate above the tube plate be less in thickness than the tube plate, its thickness should be used in the formula; that is, the smallest of the products $r \times T$ should be used.

Put in the head sheets and make all the joints of the shell. Project the furnace and combustion chambers and lay off the corrugations and laps of the furnace connections. Project the tubes across and put in all screw stays both in the longitudinal and transverse views.

Curved Tops of Combustion Chambers.—The combustion chambers are to have rounded tops, opposite ones to be tied together by gusset stays, and the rounded parts to be further stiffened by angle irons.

On account of the want of continuity, the curved tops of the combustion chambers are not capable of standing the same pressure as a complete cylinder of the same radius, and should be stiffened by angle or T bars. The constants for plain cylindrical furnaces should be considerably reduced when applied to curved tops, as the tops have been known to come down at about one-half the pressure suitable for a round furnace of the same plate thickness and radius. The curved tops should be efficiently stayed to the ends of the boilers, or with double combustion chambers back to back they should be efficiently tied together. The breadth of the combustion chamber in inches multiplied by the radius of curvature in inches is the least surface for which stay power is required. The radius of curvature should not be less than the width of the chamber, so as to avoid flat surfaces on the top and near the tube sheet. The bottoms of the combustion chambers are also stiffened with angle irons.

For determining the size of girders which support the flat top of the combustion chamber, the girder is treated as a beam supported at the ends and loaded at three points, as usually each girder carries three stay bolts. The formula for this is from Church's Mechanic's, page 270, and is:—

$$\frac{f \times I}{y} = P_0 l_2 - P_1 (l_2 - l_1), \text{ where,}$$

$f = 8{,}500$ lbs. safe stress in outer fiber of girder.

$I =$ moment of inertia of section $= \frac{bd^3}{12}$.

$y = \frac{d}{2}$, distance of outer fiber from neutral axis.

$P_0 =$ load at one end of girder.
$P_1 =$ load on one girder stay bolt.
$l_1 =$ one pitch of girder stay bolts.
$l_2 =$ two pitches of girder stay bolts.
$b =$ breadth of girder $= 1\frac{3}{4}''$ in this case.
$d =$ depth of girder.
Length of girder = depth of combustion chamber.
The girders are arranged in pairs, so b is the breadth of the pair.

Head Braces.—The area supported by each brace by measurement on the drawings is 15″ × 14.5″.

$$A = \frac{15 \times 14.5 \times 160}{9{,}000} = 3.87 \text{ sq. in.}$$

d = 2″.25 at bottom of thread.

Diameter of washer = 6″.

Thickness of washer = ¾″.

Braces over Center Furnace.—Area supported by each about 15″ × 10″.

$$\text{Area brace} = \frac{15 \times 10 \times 160}{9{,}000} = 2.67 \text{ sq. in.}$$

∴ d = 1⅞″.

Nuts, threads, etc., from p. I.

Stays below Furnaces.—The positions of these braces are fixed by the relative positions of the shell, furnace flues and manholes.

Having fixed their positions as at X Y Z, Fig. 7, the braces may be proportioned by supposing them to support the triangle (X Y Z) formed by joining them, together with half the area included between the sides of this triangle and the adjacent shell and flues. To approximate to this area, erect perpendiculars at the middle points of the upper sides of the triangle. Take B, C, and E half way between the sides of the triangle and the circumference of flues, etc. Draw A B G, G E F, and A C F, so that these lines divide equally the area between the sides of the triangle, X Y Z, and the flues or shell. The braces are calculated so that X supports A B D C, Y supports C D E F and Z supports B D E G.

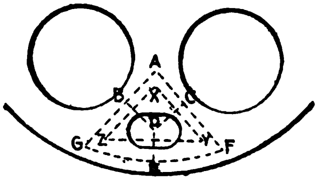


Fig. 7.

For the present case:—

Area supported by brace X = 128 sq. in.

$$\text{Area of brace} = \frac{128 \times 160}{9{,}000} = 2.27 \text{ sq. in.}$$

∴ diameter = 1″.7.

Area supported by stays, Y and Z = 120 sq. in.

Area of stay = 2.13 sq. in. ∴ diameter = 1⅝″.

Of these three stays the upper one will be a through stay passing between the combustion chambers to the opposite head; the lower stays will be diagonal and secured to the sides of the shell by "palm ends" and rivets.

Diagonal stays are never used if a horizontal stay can be used, as the resultant tension is greater on the diagonal stay.

If a = area of diagonal stay,

$$a = \frac{\text{area supported in sq. inches} \times \text{pressure}}{f} \times \text{sec. A.}$$

Where A is the angle the diagonal stay makes with the bottom of the boiler, in the same radial plane as the stay.

The braces around other manholes are direct stays fastened to the head and to the front sheets of the combustion chambers.

The washers used for main stays are riveted to the head.

Screw Stays.—The combustion chambers are stayed to each other and to the shell of the boiler by stays screwed into both sheets and fitted with nuts; the nuts to be set up on beveled washers where the stays do not come square with the plates. The holes for screw stays will be tapped in both sheets while they are in place.

All screw stays and braces will have raised threads; all braces will be made without welds.

Pitch of stays = 7″.

$$\text{Area of stay} = \frac{7 \times 7 \times 160}{6{,}000} = 1.306 \text{ sq. in.}$$

Diameter = 1″.25 or $1\frac{1}{2}$ over threads.

For Pitch of Stays and Braces.—The pitch, l, of the stay bolts and braces depends in a measure on the thickness of the flat plates supported, as will appear by treating the square surface supported as a beam supported at the ends and uniformly loaded. The formula for this is:—

$$\frac{f \times I}{y} = \frac{W \times l}{12}, \text{ where } f = \frac{65{,}000}{4.5}, I = \frac{l \times t^3}{12}, y = \frac{t}{2}, W = p \times l^2.$$

The area supported by each stay or brace is l^2. If p is the pressure per sq. inch on the flat surface, then the total pressure supported by each stay or bolt will equal $p \times l^2$. The safe load per sq. inch of section of good steel stays and braces is 8,000 lbs., therefore the area of each stay must be such that:—

$$8{,}000 \times \frac{\pi d^2}{4} = p \times l^2.$$

The diameter d of the stays is taken at the root of the threads, where the ends are fitted with nuts.

Grate Bars are to be of wrought iron, so fitted as to be worked under forced draught without opening the doors.

Bridge Walls are of cast iron; to be readily removable, finished with fire brick.

Furnace Fronts are to be made with double walls of wrought iron, the space between the walls to be in communication with the ash pits. The upper part of inner plate is to be perforated as directed; dead-plate to be of cast iron and easily removable.

Furnace Doors must be protected in an approved manner from the heat of the fire.

Ash Pit Doors to be made of $\frac{1}{8}''$ iron, stiffened with angle irons. False ash pans of $\frac{1}{4}''$ iron are to be fitted under all grates.

Circulating Plates are to be fitted at each side of each nest of tubes, made so as to be readily removed and introduced through manholes.

The *Uptakes* are to be of wrought iron, No. 8 B. W. G., built on channel irons, stiffened with angle irons, and bolted to the boiler heads and shells. Outside of the uptake is to be a jacket enclosing a three-inch air space. This space is to be filled with an approved non-conducting material.

Uptake Doors are to be double shells of iron, No. 8 B. W. G. The space between the shells is to be fitted with the same non-conducting material as used for the uptakes.

Smoke Pipes are to be two in number; the top about 60 feet above the grates, the bottom part shaped to join the uptakes. The top is to be finished with angle irons and with a hood covering the casing. The pipes are to be stayed by shackles and guys with turnbuckles for setting up the guys.

A casing enclosing a three-inch air space extends from the uptake to about six inches below the hood at the top. An additional casing is placed for five feet or more above smoke pipe hatches.

Boiler Attachments are to be supplied as follows:—

1 self-closing main steam stop valve.

1 self-closing auxiliary steam stop valve.

1 dry pipe.

1 main feed check valve with internal pipe.

1 auxiliary feed check valve with internal pipe.

1 bottom blow valve with internal pipe.

1 surface blow valve with internal pipe and scum pan.

2 safety valves to be connected with dry pipes, or to have internal pipes.

1 steam gauge, on single-ended; 2 on double-ended boiler.

2 glass gauges with automatic cocks on single-ended, and 3 on double-ended boiler—two of them being at feeding end.

4 gauge cocks on each end of double-ended boiler.

1 sentinel valve of ½″ area.

1 salinometer pot.

1 drain cock.

1 air cock.

1 approved circulating apparatus.

1 cock threaded for the attachment of a syringe.

All external fittings, unless specially stated to the contrary, are to be of composition.

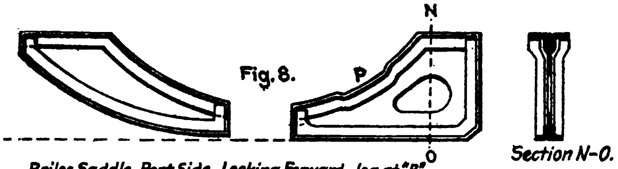
All fittings are to be flanged and through bolted.

All cocks, valves and pipes are to have spigots or nipples passing through the boiler plates.

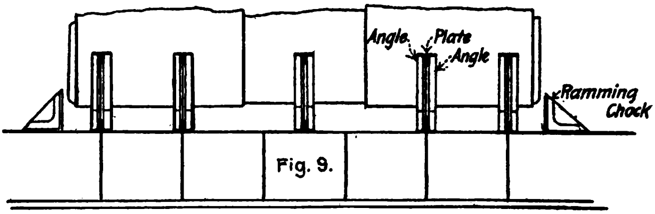
Internal pipes of brass must not touch the boiler plates except where they join their external fittings.

The stems of all valves on the boilers are to have outside screw threads.

Securing Boilers in Iron and Steel Ships.—The boilers are generally placed on board the ship after launching, as the vessel being waterborne is less liable to be strained from the concentration of weight. The framing of the boiler hatches should be so arranged as to give ample room for lowering the boilers into their compartments. The ship is taken under the shears or derrick, the boilers, slung by straps, lifted on board and lowered on blocking. They are then moved along the floors by jacks until in their proper positions over the saddles and are then lowered into place. The boilers rest on saddles and are secured to them by bolts. The number of saddles depends on the size and kind of boilers used, being generally two for single-ended and four or five for double-ended boilers. They should be spaced so as to allow access to the circumferential seams for caulking. Where double-ended boilers are used, the fire rooms are generally athwartship. The saddles are then arranged so as to come directly over the frames. Each saddle consists of a vertical plate secured to the inner bottom and reverse frames by angles. The top of this plate is cut to the curvature of the boiler, to which it is secured by angles on each side, jogged if necesary to fit over the longitudinal butt straps. The boiler rests on the angles. The boilers are screwed to the saddles by bolts and nuts. The bolts are screwed through the

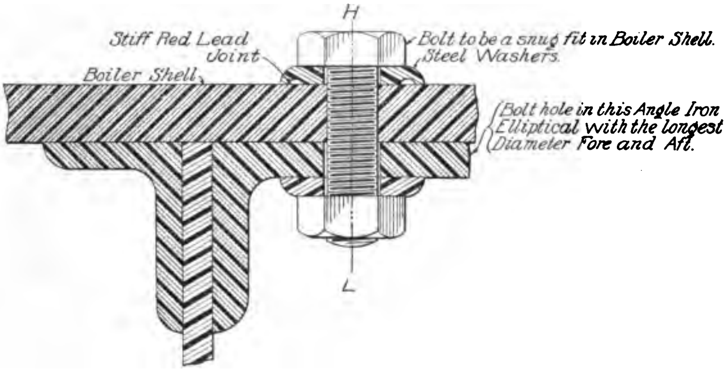


Boiler Saddle, Port Side, Looking Forward, Jog at "P" for Butt-Strap.



Showing Saddles in Position.

shell from the inside and have a cup washer under the head, the joint being made tight by red lead putty under the washers. The holes through the flanges of the saddles are elongated in a fore and aft direction to allow for expansion. In addition to the above, some means must be adopted to prevent displacement in a fore and aft direction, if the vessel is intended to be used for ramming. The usual method is to fit what are called ramming chocks. These consist of a vertical plate about one inch thick fitted to each end of the boiler and secured to the inner bottom by angles. They are placed opposite the lowest point of the boiler and are fitted so as to extend about six inches above the lowest point and a clearance of about one-quarter inch is left between them and the boiler. In large boilers two are used at each end. When the boilers are placed with their axes athwartships the saddles should be placed over the longitudinals; when this is not possible, heavier plates should be worked in the inner bottom below the saddles. The saddles and the methods of securing the boilers to them are illustrated in the Figs. 8, 9 and 10.


H
Stiff Red Lead Joint
Boiler Shell
Bolt to be a snug fit in Boiler Shell.
Steel Washers.
Bolt hole in this Angle Iron Elliptical with the longest Diameter Fore and Aft.
L

RIVETING.

FORMULÆ FOR PROPORTIONING RIVETED JOINTS AND CALCULATING THE WIDTH AND THICKNESS OF BUTT STRAPS FOR STEEL BOILERS.

With the aid of these formulæ the tables for riveting joints in Traill's work are calculated, but for every day work in the designing room it will undoubtedly be a time-saving operation to obtain the desired information direct from those tables. For convenience in comparing with the above-mentioned tables the same nomenclature is used. The following assumptions are made:

First:—That the mean tensile strength of the plate is 28 tons and the mean shearing strength of the rivets is 23 tons per square inch of net section.

Second:—That rivets in double shear offer 1.75 times the resistance to shearing exerted by rivets in single shear.

Let p = pitch of rivets in inches (greatest).
d = diameter of rivets in inches.
c = constant whose value is 1. for lap or single butt strap, and 1.75 for double butt-strapped joints.
A = area of one rivet in square inches.
n = number of rivets in the greatest pitch.
$\%$ = percentage of plate left between rivets in the greatest pitch.
$\%_1$ = percentage of the rivet section compared with that of the solid plate.
$\%_2$ = percentage of the combined plate and rivet section when alternate rivets are omitted in the row.
p_d = the diagonal pitch.
E = the distance from the center of the rivets to the edge of the plate.
V = the distance between the rows of rivets in ordinary zigzag riveting, and in chain riveting when alternate rivets are omitted in the outer row.
V_1 = the distance between the outer and the next row of rivets in zigzag riveting when alternate rivets are omitted in the outer row.
t = the thickness of the plate in inches.
t_1 = thickness of a single butt strap in inches.
t_2 = the thickness of a double butt strap in inches.

TO FIND THE PERCENTAGE STRENGTH OF ANY GIVEN JOINT.

The metal left between the rivet holes = (p — d) t, and the percentage strength of this as compared with the solid plate is:—

$$\frac{100\,(p - d)}{pt} \times t. \quad \therefore \% = \frac{100\,(p - d)}{p} \quad . \quad . \quad . \quad . \quad . \quad (1)$$

The resistance offered to shearing by the rivets in a single pitch is $23 \times A \times n \times c$; comparing this with the strength of the solid plate = $28 \times p \times t$, we have:—

$$\%_1 = \frac{100 \times 23 \times A \times n \times c}{28 \times p \times t} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

Considering now the case where alternate rivets are omitted in the outer row, in the next row there will remain $(p - 2d) \times t$ solid metal, and $\frac{100\,(p - 2d)}{p}$ will be the percentage strength of this row as compared with the solid plate; but, supposing that the plate tears along this row of rivets (see Fig. 12, p. 122), before a total giving away of the joint occurs one rivet for each pitch must also be sheared in the outer row and the percentage strength of this single rivet is $\frac{1}{n}$, which value will be obtained from equation (2). From the above considerations we have

$$\%_2 = \frac{100\,(p - d)}{p} + \frac{1}{n} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

The lowest of the values obtained from equations (1), (2) and (3) is the percentage strength of the joint. An examination of these equations shows that for double butt-strapped joints, so long as the diameter of the rivets is not less than the thickness of the plates, $\%_2$ is always greater than $\%$ or $\%_1$. This is also the case with lap joints, so long as the diameter is not less than

$$\frac{t}{\frac{23}{28} \times \frac{3.1416}{4}} = \frac{t}{.64515},$$ and, as both these conditions usually hold, the use of formula (3) will seldom be necessary.

To find d, when p, c, n, and t are given, so that $\%$ may equal $\%_1$ (that is, so that plate and rivet section may be equivalent in strength).

From (1) and (2) we have:—

$$\frac{100\,(p - d)}{p} = \frac{100 \times 23 \times A \times n \times c}{28 \times p \times t} \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

Substituting for A its value $= \frac{3.1416 \times d^2}{4}$ and simplifying, we have:—

$$d^2 + \frac{1.55\,td}{n \times c} = \frac{1.55\,pt}{n \times c}.$$

Solving for d we have:—

$$d = \sqrt{\left(\frac{.775\,t}{n \times c}\right)^2 + \frac{1.55\,pt}{n \times c}} - \frac{.775\,t}{n \times c}$$

which reduces to:—

$$d = \sqrt{\frac{.775\,t}{n \times c}\left(\frac{.775\,t}{n \times c} + 2\,p\right)} - \frac{.775\,t}{n \times c} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

If d, c, n and t are given; from equation (4) we have:—

$$p = \frac{23 \times A \times n \times c}{28 \times t} + d \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (6)$$

To find d and p, when n, c and t and $\% = \%_1$ are known:—

From (1) we have $p = \frac{100\,d}{100 - \%}$ . . . . (7)

Substituting this value for p and letting $A = \dfrac{22\,d^2}{7 \times 4}$ in (2),

we have:— $$\% = \%_1 = \frac{100 \times 23 \times 22 \times d^2 \times n \times c}{28 \times 7 \times 4 \times \dfrac{100\,d}{100 - \%} \times t}$$

Solving:— $$d = \frac{1.55 \times \% \times t}{(100 - \%) \times n \times c} \quad . \; . \; . \; . \; . \; . \; . \quad (8)$$

Substituting this value of d in (7) we have:—

$$p = \frac{100 \times 1.55 \times \% \times t}{(100 - \%)^2 \times n \times c} \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad (9)$$

It will generally be found simpler to substitute the value for d found from equation (8) in equation (7), thus obtaining a value for p direct, without using equation (9).

DIAGONAL PITCHES AND THE WIDTH OF BUTT STRAPS.

The value of the metal in the diagonal pitch p_d is only about $\frac{5}{6}$ of what it is in the horizontal pitch, or in that part of the horizontal pitch to which it is required to be equivalent in strength. In any case, the diagonal pitch should not be less than that found by the following formulæ:—

I.

Ordinary zigzag riveting and chain riveting with alternate rivets omitted in the outer row.

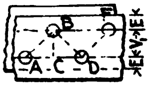


Fig. 11.

Reference to figure (11) shows that the same reasoning applies to both cases, as in each case the net section of two diagonal pitches must be made equivalent in strength to the net section contained in the greatest horizontal pitch. The liability of the sheet to tear along A B and B D would be the same as along A D or B F.

Now the metal left along A B D is $2\ (p_d - d)\ t$ and along A D or B F is $(p - d)\ t$.

$$\therefore 2\ (p_d - d)\ t = \tfrac{6}{5}\ (p - d)\ t, \text{ and } p_d = \frac{6\,p + 4\,d}{10} \quad . \quad (10)$$

From the right angle A B C we have for the distance between the rows of rivets:—

$$V = \sqrt{p_d^2 - \left(\frac{p}{2}\right)^2} = \sqrt{\left(\frac{6\,p + 4\,d}{10}\right)^2 - \frac{p^2}{4}} = \sqrt{\frac{11\,p^2 + 48\,pd + 16\,d^2}{100}}$$

$$\text{or } V = \frac{\sqrt{(11\,p + 4\,d)\,(p + 4\,d)}}{10} \quad . \; . \; . \; . \; . \; . \; . \; . \quad (11)$$

The authors mentioned before, state that, for chain riveting, this distance should not be less than $\frac{4\,d + 1}{2}$ and, as this result is greater than that obtained from (11), it is the one that is usually found tabulated.

The distance E from center of rivet to edge of plate should not be less than 1.5 d; so that the minimum lap of sheets, if lap joint, or half width of butt strap $= 2\,E + V \quad . \; . \; . \; . \quad (12)$

II.

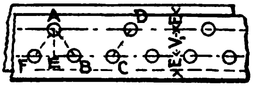


Fig. 12.

Zigzag riveting with alternate rivets omitted in the outer row.

An examination of figure (12) shows that rupture may occur by the plate giving away along A B C D, or along A D, and to make the joint symmetrical in strength, the net section along A B C D must be equivalent in strength to the net section along A D. Now $B\,C = \frac{p}{2}$ and the net section along B C is $\left(\frac{p}{2} - d\right)$ t, while on A D it is $(p - d)\ t$; hence the section to which the metal left on the lines A B and C D must be equivalent in strength is:—

$(p - d)\ t - \left(\frac{p}{2} - d\right) t = \frac{p}{2}\ t$, but the net section on A B or

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C D is $(p_d - d)\ t$; therefore, $2\ (p_d - d)\ t = \frac{6}{5}\left(\frac{p}{2}\ t\right)$, and

$$p_d = \frac{3\,p + 10\,d}{10} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (13)$$

In the triangle A F E we have $F\,E = \frac{B\,F}{2} = \frac{p}{4}$, and, for the distance between the rows of rivets, we have:—

$$V_1 = \sqrt{\left(\frac{3\,p + 10\,d}{10}\right)^2 - \left(\frac{p}{4}\right)^2} = \sqrt{\frac{9\,p^2 + 60\,pd + 100\,d^2}{100} - \frac{p^2}{16}}$$

$$= \sqrt{\frac{11\,p^2 + 240\,pd + 400\,d^2}{400}} = \frac{\sqrt{(11\,p + 20\,d)\,(p + 20\,d)}}{20} \quad . \; . \; (14)$$

As before, the half breadth of butt strap $= 2\,E + V_1$ . . (15)

As the riveting in all ordinary cases is either of the form shown in Fig. 11 or Fig. 12, or a combination of those forms, equations (10), (11), (12), (13), and (14) have a general application. For instance, consider the case of a double butt-strapped, treble riveted, zigzag joint with alternate rivets omitted in the outer row; the distance V_1 between the outer and second row is obtained from equation (14), the distance V is obtained from equation (11). In this case the half breadth of butt strap $= 2\,E + V_1 + V$ . . (16)

THICKNESS OF BUTT STRAPS.

From considerations of strength and to secure tight joints the aggregate thickness of the butt straps should always be more than that of the plate. For single butt straps $t_1 = \frac{9}{8}\ t$; for double butt straps $t_2 = \frac{5}{8}\ t$, are arbitrarily taken as the minimum values allowed.

When alternate rivets are omitted in the outer row, the thickness obtained above must be increased.

The ratio of this increase is expressed by $\frac{p - d}{p - 2\,d}$, which is the ratio of the net section left in the outer row to the net section left in the next row. With this form of riveting, the minimum values of t_1 and t_2 will be found by the following equations:

$$t_1 = \frac{9}{8}\left(\frac{p - d}{p - 2d}\right) t. \qquad t_2 = \frac{5}{8}\left(\frac{p - d}{p - 2d}\right) t.$$

BOILER PROBLEM.

WORKING SHEET.

Required:—A set of boilers to supply steam of........lbs. pressure to......................engines, such as are used in a modern naval vessel of the...................class, when developing under an air pressure of..........inches of water, a maximum of..........I. H. P for............consecutive hours.

Additional specifications ....

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1. Name of ship..

2. Maximum I. H. P...

3. Boiler pressure, pounds per gauge.....................................

4. Air pressure, in inches of water......................................

5. Coal burned per sq. ft. of grate, assumed from trials.................

6. Coal burned per I. H. P., assumed from trials.........................

7. I. H. P. per sq. ft. of grate, assumed................................

8. Pounds of water per I. H. P. per hour, assumed........................

9. Water evaporated per pound of coal, assumed...........................

RESULTS OF PRELIMINARY CALCULATIONS.

10. Grate surface, from (2) and (7)......................................

11. Grate surface from (2), (5) and (6)..................................

12. Grate surface from (2), (5), (7) and (8).............................

13. Grate surface used...

14. Trial diameter of inside of furnace..................................

15. Trial length of grate..

16. Number of furnaces, from (13), (14) and (15).........................

17. Number of furnaces used..

18. Type of boiler as determined from tracing of boiler space in ship

....

19. Limiting diameter of boiler as determined from tracing..........

20. Number of furnaces in boiler head.............................

21. Length of bridge wall, assumed.................................

22. Length of hanging bridge wall.................................

23. Beginning of grate from outside of front tube-sheet............

24. Length of combustion chamber.................................

25. Distance between backs of same, to p........bottom.............

26. Approximate length of boiler...................................

27. Length of tubes..

28. Length of tubes for low boiler, assumed.........................

29. Volume of boiler per I. H. P., assumed..........................

30. Diameter of boiler...

31. Numbers of boilers and class...................................

32. Thickness of shell, Bureau method.............................

33. Thickness of shell, Board of Supervising Inspectors' rule........

34. Thickness of shell, Lloyd's rules...............................

35. Thickness of shell used...

36. Ratio of corrugated length of flue to axial length................

37. Heating surface of furnace......................................

38. Heating surface of combustion chamber, from planimeter measurements of rough drawing of combustion chamber...........

39. Total heating surface, Bureau method..........................

40. Total heating surface, Seaton...................................

41. Total heating surface..............method....................

42. Total heating surface from above methods.....................

43. Tube heating surface...................................

44. Calorimeter ....

45. Number of tubes in each boiler..............................

46. Number of tubes in each end, by calculation....................

47. Number of tubes in each end, from rough drawing.............

Having completed all of the foregoing preliminary calculations, make a rough drawing, to as large a scale as the paper will admit, determine all the details, and tabulate the results in following columns before commencing the finished drawing on the board.

FINAL DATA OF ONE..........ENDED BOILER, ASSEMBLED TO COMMENCE DRAWING.

48. Diameter of boiler...

49. Radius of arc of furnace centers...............................

50. Distance between furnace centers measured on chords...........

51. Horizontal distance of middle furnace centers from center of head of boiler ....

52. Number of furnaces in head.................................

53. Inside diameter of furnace..................................

54. Outside diameter of furnace.................................

55. Thickness of furnace ....

56. Thickness of shell plates....................................

57. Thickness of front tube sheet..................................

58. Thickness of back tube sheet..................................

59. Thickness of top head sheet..................................

60. Thickness of lower head or furnace sheet......................

61. Thickness of combustion chamber sheets........................

62. Thickness of butt straps......................................

63. Distance of top of top row of tubes from top of boiler...........

64. Water space between shell and combustion chamber, at top......
.........................., at bottom ....

65. Water space between shell and nearest furnace..................

66. Water space between two adjacent combustion chambers.........

67. Water space, minimum, between nests of tubes..................

68. Number of combustion chambers in each end...................

69. Tubes plain, number in one end..............................

70. Tubes stay, No........B. W. G., outside diameter..............

71. Tubes stay, number in one end...............................

72. Tube spacing, all, horizontally............vertically............

73. Total number of tubes in one end.............................

74. Length between tube sheets...................................

RIVETED JOINTS.

75. Back end of furnaces to combustion chambers, and to adjacent
furnaces.....................diameter of rivets..............
pitch...................lap of joint.........................

76. Upper head sheet to front tube sheet...........................
diameter of rivets..........., pitch..........., lap..........

77. Lower edge of tube sheet to furnace sheet......................
diameter of rivets..........., pitch..........., lap...........

78. Middle circumferential joint......................................

diameter of rivets..........., pitch..........., lap............

79. End circumferential seams ....

diameter of rivets..........., pitch..........., lap............

80. Butt strap............, width............, thickness............

81. Butt joint, kind of joint.....................................

diameter of rivets..........., pitch..........., lap............

distance between rows of rivets...............................

82. Length of boiler..

83. Length of one ring..

84. Radius of curved head...

85. Braces above tubes, diameter..................................

86. Braces to back tube sheet, diameter...........................

87. Braces above lower manholes, diameter.........................

88. Braces, diagonal, in lower water space........................

89. Screw stays, diameter................, pitch....................

90. Heating surface, plate, square feet...........................

91. Heating surface, tube, square feet............................

92. Heating surface, total, square feet...........................

93. Length of grate ....

94. Grate area, one boiler..

95. Ratio of heating surface to grate area.........................

96. Volume of steam space, water 6″ above tubes...................

97. Water surface, square feet....................................

98. Area through tubes, square feet...............................

99. **Area over bridge-wall, square feet**..............................

100. **Volume of furnaces and combustion chambers above grates**......

101. **Combustion chamber head, curved or flat**......................

102. **Combustion chamber bracing**.................................

103. ....

104. ....

105. ....

SCREW PROPELLERS.

DATA GIVEN.

Name of ship.
Number of screws.
Number of blades.
Kind of screw.
Adjustment of pitch, each side.
Displacement (tons).
Draught, forward.
Draught, aft.
Draught, mean.
Total I. H. P.
Revolutions per minute.
Speed in knots.
Diameter of screw shaft.
Thickness of sleeve on shaft.
Metal of screw.
Working face moved aft on center line.
Blades bent back, or center line bent aft.
Scale of drawing.

(For calculating the diameter by comparison, use the propeller of the U. S. Steamer "Olympia," $d = 14' 9''$, $p = 13{,}500$, $v = 20$ knots, $r = 129$.)

NOTES ON PROPELLER DESIGN.

The following notes are compiled from:

Seaton: "A Manual of Marine Engineering."
Seaton and Rounthwaite: "Pocket-book of Marine Engineering."
Chase: "Screw Propellers and Marine Propulsion."
Barnaby: "Marine Propellers."
Foley: "Mechanical Engineer's Reference Book."
Practice of the Bureau of Steam Engineering of the Navy Department.

In all cases, the Department method is given preference.

DEFINITIONS.

Chase, p. 132: The diameter of a screw propeller is the diameter of a circle swept by the tip of the blades.

The pitch is the axial distance between the same thread at one revolution.

The disc area is the area of the circle described by the tips of the blades less the area of the hub. (The area of the hub is included by some authorities.)

The developed or helicoidal area is the actual area of the working face of the blades when flattened out on a plane.

The projected area is the area of the projections of the blade on a plane taken at right angles to the axis of the screw.

The length of the screw is the greatest length of the blade measured parallel to the axis.

The speed of the screw is the pitch multiplied by the number of revolutions.

The pitch ratio is the quotient arising from dividing the pitch by the diameter.

A screw propeller is a true screw when its pitch is uniform; and an expanding pitch screw when the pitch increases from the forward to the after edge, or radially, or both.

The principal feature of the Griffiths screw is the large boss, which, while not impairing the efficiency, enables the blade to be fixed in such a manner that the pitch can be readily altered.

The modified Griffiths screw propeller is generally adopted for vessels of the navies of England and the United States, and is one having blades of any curved shape diminishing in width towards the tips and having the broadest part nearer the boss than the tips.

Chase, p. 139: "The shape of the blade is sometimes oval and sometimes resembles the section exposed by cutting a pear longitudinally. When made pear-shaped, it is for the purpose of getting the greatest possible surface near where the pitch angle is 45°, as that is considered the most efficient angle; and by the blade becoming diminished in width towards the tips, the surface friction is thereby reduced, and room is obtained for the access of the supply water."

CALCULATIONS.

Chase: "In commencing the design of a propeller for a given ship, the diameter and pitch are first decided upon."

Chase, p. 130: "The true secret of success in designing a screw propeller lies in the correct proportion of the three cardinal elements—diameter, pitch and blade surface; and that the shape of the blades (within certain limits), their thickness, and the condition of their surfaces, are of minor importance."

PITCH.

Seaton, p. 329: "Let S = speed of ship in knots; R = number of revolutions per minute; s = slip in knots, generally expressed as a percentage, x. Then S = speed of screw $\left(1 - \frac{x}{100}\right)$.

$$\text{Speed of screw} = \frac{S}{1 - \frac{x}{100}} = \frac{S \times 100}{100 - x}.$$

$$\text{Then pitch} = \frac{S \times 100 \times 6{,}080}{60\,R\,(100 - x)} = \frac{S}{R} \times \frac{10{,}133}{100 - x}.$$

Chase, p. 134: "The percentage of slip now allowed is from sixteen to twenty in large screws for good-sized ships, in which the engines make from 115 to 150 revolutions per minute."

Seaton: "x is generally from 20 to 25% for naval vessels."

From 24 of the latest fast vessels of the United States with engines of Bureau design, the average slip is 20.77%.

Seaton, p. 323: "If the efficiency of the mechanism were the same at any number of revolutions, a fine pitch propeller might be used to advantage, as being more efficient than a coarse one; but since the efficiency of the engine is much higher at a moderate than at a maximum rate of speed, the total efficiency of the screw and engine is often improved by increasing the pitch of the screw."

PITCH RATIO.

Chase, p. 137: "For torpedo-boats and other light-draught boats, which usually run in smooth water and where high-rotative velocities prevail, diameters being restricted in consequence of small draught, the pitch of screws varies from about 1.4 to two times the diameter. For the larger high-speed ocean vessels, which traverse water more or less rough and where ample diameter and immersion can be obtained, the pitch varies from 1.2 to 1.6 times the diameter. For slower cargo vessels, etc., the pitch varies from 1.1 to 1.3 times the diameter."

Foley: "As a rule the pitch ratio should lie between 1 and 1.5 times the diameter; 1.25 times the diameter is usually satisfactory in practice."

From 24 vessels of the U. S. Navy with engines of the latest Bureau design, the pitch averages 1.23 times the diameter.

EXPANDING PITCH.

Chase, p. 32: "The alleged advantages for a propeller of expanding pitch are that, as the screw advances through the water, the forward portion encounters a resistance due to a solid body moving through water at rest; but water, being an exceedingly mobile substance, that with which the screw first comes in contact has motion imparted to it by the pressure exerted by the screw; and this motion of the water is accelerated by the portions of the blade following, and consequently the after-portions of the screw will press upon the water with a motion opposite to that in which the screw is advancing. In order, then, to equalize the pressure over the whole blade, the pitch must be increased proportionately to the motion acquired by the water. The increase in pitch usually employed is from 15 to 25%."

It is questionable whether there is any advantage in expanding pitch.

DIAMETER.

Seaton, p. 323: "The size of a screw depends on so many things that it is very difficult to lay down any rule for guidance and much must always be left to the designer, so as to allow for all the circumstances in each case."

The diameter of the propeller must be less than the draught of the ship, to prevent racing and striking ground.

Barnaby, p. 34: "It is very important that a propeller should have sufficient immersion, since, if it breaks the surface of the water, the efficiency is reduced to a remarkable extent; but, if it is sufficiently far below the surface to prevent its drawing air, any further immersion within the limits that can practically be obtained is of little value."

Chase, p. 140: "The depth of immersion of the upper tips of the blades should increase with the draught of the vessel or the diameter of the propeller. It should not be less than in the following table when the vessel is at the load line:

For a diameter of	4 ft. or less..........	$\frac{1}{11}$	of diameter.
" " "	4 to 8 ft...........	$\frac{1}{10}$	" "
" " "	8 to 12 ft...........	$\frac{1}{9}$	" "
" " "	12 to 16 ft...........	$\frac{1}{8}$	" "
" " "	16 to 20 ft...........	$\frac{1}{7}$	" "
" " "	20 ft. or over.........	$\frac{1}{6}$	" "

From practice: For large vessels there should be from 1½ to 2 feet of space from the bottom of the keel to the bottom of the blades, and from 4 to 4½ feet from the surface of the water to the blades.

Seaton, p. 323: "The diameter of the screw must bear relation to the turning power of the engine; for, if made of too small dimensions, it cannot absorb the power of the engine, however coarse the pitch may be."

Since the area of surface on which the skin resistance acts increases as the square of the distance from the center, and the arm on which it revolves increases with the distance from the center, the frictional resistance per sq. ft. of surface varies as the cube of the distance from the center, so that any increase in diameter means a large increase in resistance.

If a screw is of sufficient diameter and has ample blade area, any addition beyond the tip will act then as a brake and seriously reduce its efficiency.

Seaton, p. 324: "Diameter of screw $= K \sqrt{\frac{I.\,H.\,P.}{(P \times R)^3}}$

$I.\,H.\,P. =$ total — 6 to 10 per cent. for auxiliaries. $P =$ pitch in feet. $R =$ revolutions per minute. $K =$ constant." Seaton gives for K a value of 20,000.

From trials of similar ships the value of K is determined for the one being designed.

Seaton and Rounthwaite give a value of 22,700 for naval vessels of fine lines.

From 23 vessels of the U. S. Navy with engines of the latest Bureau design, the average value of K is 22,908.

Foley: "$D = \sqrt{\frac{I.\,H.\,P. \times C}{S^3}}$. $S =$ speed in knots. $I.\,H.\,P. =$ total I. H. P. less percentage used for auxiliaries. $C = 212$ for speeds from 8½ to 9 knots and 258 for speeds from 15 knots upwards. Also, $D =$

$$\sqrt[3]{\frac{I.\,H.\,P. \times K}{\text{Rev. Per Min.} \times S^2}} \qquad \text{where } K = \begin{cases} 30{,}000 \text{ if } P = D. \\ 24{,}000 \quad \text{"} \quad = 1.25\,D. \\ 20{,}000 \quad \text{"} \quad = 1.50\,D.\text{"} \end{cases}$$

PERIPHERAL SPEED.

Foley: "Owing to the great loss of efficiency from friction at high speeds, it is usual not to exceed about 6,500 ft. per minute at periphery."

This should be applied as a check after the diameter is calculated.

DIAMETER BY COMPARISON.

Barnaby, p. 69: "To find the diameter of a propeller for a given I. H. P. and a given speed from the diameter of another similar propeller at a different I. H. P. and at different speed:

If d = diameter of model, which may be larger or smaller than D;
D = diameter of required propeller;
p = I. H. P. model;
P = I. H. P. of required propeller;
v = speed of vessel with model propeller;
V = " " " " required propeller;
r = revolutions of model propeller;
R = " " required propeller.

Then $D = \sqrt{d^2 \times \frac{v^3}{V^3} \times \frac{P}{p}}$ and $R = r \times \frac{V}{v} \times \frac{d}{D}$.

The pitch ratio must be the same as that of the screw, which is treated as a model."

AREA.

Seaton, p. 329: "The best area is not easily determined, except by experiment by data derived from the performance of similar ships with similar screws.

Good results are obtained by:

$$\text{Total area of screw blades} = K\sqrt{\frac{\text{I. H. P.}}{R}}.$$

	Single Screws.	Twin Screws.
The value of K for four bladed is..	15	10.5
" " three " ..	13	9.0
" " two " ..	10	

Chase, p. 158: "For naval vessels with twin screws K = 7 to 8.5."

Seaton and Rounthwaite: "For very fine-lined naval vessels with three-bladed screws K = 7.75."

From 21 vessels of the U. S. Navy with engines of latest Bureau designs, K = 8.847.

Seaton, p. 330: "The projected area is often taken as the criterion, and is to some extent a true one."

AREA FROM DISC AREA.

Foley: "The developed surface is usually from 28 to 40 per cent. of the disc area. For ordinary cases of moderate or high speeds, where the diameter is well proportioned, 35 per cent. is usually satisfactory. It should be remembered that there may be great loss of power from too much surface."

For the disc area the boss is usually spherical, and from $\frac{1}{4}$ the diameter of the screw for small screws to $\frac{1}{5}$ the diameter for large ones.

AREA FROM THRUST.

Seaton, p. 201: "$P = \text{I. H. P.} \times \frac{217}{K}$. P is mean normal thrust; K is speed in knots per hour."

"P varies with the I. H. P. and inversely as the speed in knots, so the thrust may vary considerably. If the speed is from any cause reduced the thrust must increase."

"To allow for the variations of P, the surface should be such that the pressure per sq. in. from the mean normal thrust does not exceed 70 lbs."

$$\text{Indicated thrust} = \frac{\text{I. H. P.} \times 33{,}000}{\text{Revs. per min.} \times \text{pitch in feet}}.$$

"The efficiency of the engine and propeller are each taken at 75%.

Then, I. H. P. × 33,000 becomes I. H. P. × 33,000 × .75 × .75 = I. H. P. × 18,562."

From six of the vessels of the U. S. Navy with engines of the latest Bureau design, the average thrust per sq. ft. of helicoidal area is 1410.17 pounds, which is much later practice than that of Seaton. This is for propellers of manganese bronze.

The area from thrust is to be a check on the other methods.

BLADES.

Chase, p. 135: "Steadiness in action determines the number of blades. For a single screw, four blades give the greatest satisfaction, especially in rough water."

"Two-bladed propellers cause too great vibration."

"With twin screws where the supply water is not so much interfered with, and where the immersion is more ample, three-bladed screws are preferable."

Seaton and Rounthwaite: "Three blades are generally used when the immersion is sufficient and the revolutions fairly high; but when the calculated diameter of the propeller becomes nearly as great as the draught of water, and the revolutions are only moderate, four blades should be used."

Seaton: "If the necessary disc area can be obtained with a three-bladed propeller, this form is preferable, as less shock is given to the ship; but if the draught is limited, a four-bladed propeller may be necessary."

Practice: Blades are generally separate from the hub in naval vessels.

MATERIAL OF BLADES.

Seaton, p. 335: "Cast-iron is generally used because cheap, and when struck breaks clean off and goes out of the way.

Cast-steel blades are largely used, being thinner and lighter for the same strength than cast iron.

Wrought-steel blades are strong and efficient. The chief objection to steel is that it corrodes rapidly on the back of the blade, due probably to air detached from the water.

Phosphor and manganese bronzes are largely employed for making propellers for naval vessels. The blades can be cast thinner than if made of steel, as there is little corrosion. They are objected to on the ground that they cause galvanic action, and injure the iron or steel near by; but this is questioned."

Seaton and Rounthwaite: "For naval vessels propellers are almost invariably of gun-metal or bronze, and except sometimes in torpedo-boats, the blades are always detachable."

Bureau Practice: "For U. S. naval vessels manganese bronze is always used in the latest designs."

FORM OF BLADES.

Seaton, p. 333: "The forms of blades are numerous and varied. Some bend the blades forward, some back; but no general plan has been found to give, always, the best results. The better results are due to a better proportion of blade rather than to the shape."

Barnaby, p. 78: "The standard blade area of the Admiralty standard blade is an ellipse of major axis equal to the radius of the propeller and of minor axis equal to four-tenths of the major axis. It is often found, however, that owing to the diameter being limited, sufficient blade area is not obtained by these proportions. In such cases the elliptical form is adhered to with an increased minor axis of from .5 to .55 of the major."

LENGTH OF BLADES.

Chase, p. 140: "This is important and is determined by practice.

"The length affects the power to move a given columnal mass of water. When made too long, the loss from the divergent and rotary direction given the discharge water and also from the surface friction of the blades more than counterbalances the advantage of the increased mass of water acted upon.

"The length should vary with the rotative speeds. The most efficient length seems to lie between 18 and 20% of the diameter—the former for high and the latter for low speeds. The average of a number of high-speed screws of modified Griffiths shape gives a maximum length of 19%."

BREADTH OF BLADES.

Seaton, p. 334: "The greatest breadth of blade should not, as a rule, be beyond one-third the radius of the disc from the center, and should be approximately as given by the following rule:

$$\text{Max. breadth of blade in inches} = K\sqrt[3]{\frac{\text{I. H. P.}}{\text{Rev.}}}.$$

"For a four-bladed screw, K = 14; for three-bladed, K = 17; and for two-bladed, K = 22.

"The breadth of blade at the tip should be from one-third to two-fifths of the maximum."

THICKNESS OF THE BLADE.

Seaton, p. 330: “The strength of the blade at the root or part near the boss will vary nearly directly as the breadth and as the square of the thickness, and for practical purposes may be considered to do so.

Let d = diameter of screw shaft close to the propeller.
b = breadth of blade in inches at a distance of 1½ d from the center of the shaft.
n = number of blades.
R = number of revolutions per minute.

Then, thickness of blade (at a distance 1½ × d from center) =

$$\sqrt{\frac{d^3}{n \times b}} \times K.$$

Also, thickness of blade (at a distance a from center) =

$$\sqrt{\frac{\text{I. H. P.} \times f}{R \times n \times b}} \times K,$$

where f may be taken as 100 for compound engines of the ordinary two- and four-cylinder type; 90 for three- and six-cylinder compound engines, and 85 for three-crank triple-compound engines.

“The value of K is 4 for cast iron; 2 for ordinary gun-metal; 1.6 for cast steel and 1.5 for forged steel and bronzes of superior make.”

Bureau Practice: “From 24 vessels of the U. S. Navy with engines of the latest Bureau design, the average value of K is 2.56.”

Chase, p. 162:

“Cast iron............	allow	.5	in. for each foot in dia. of screw.
Cast steel...........	“	.4	“ “ “ “
Gun metal..........	“	.45	“ “ “ “
High class bronze....	“	.4	“ “ “ “

THICKNESS AT TIP.

Seaton, p. 330: “The thickness of metal at the tip should be .2 of that at the root.”

Chase, p. 160: "D is the diameter of the propeller in feet.

Thickness at the tip,	cast iron	.04 D + .4 in.
" "	cast steel	.03 D + .4 in.
" "	gun metal	.03 D + .2 in.
" "	highest class bronze	.02 D + .3 in.

Also, *Chase:* "The thickness at the tip should be as small as possible consistent with good casting or forging. When gun-metal is used it is generally made about $\frac{3}{64}$ inch for each foot in diameter of the screw."

BOSS OR HUB.

Seaton, p. 332: "When a loose-bladed propeller is designed, the boss is usually spherical in general form with flats or recesses for the blades. The diameter of the sphere is from ¼ the diameter of the screw for small propellers to ⅕ the diameter for large ones."

Chase: "Boss = diameter divided by 3¼ or 3½, which will cut away about one-fifth of the standard ellipse."

Seaton, p. 332: "The length of the boss depends on the size of the base of the blades, and is generally about .85 of the diameter of the sphere for two-bladed screws and .75 for four-bladed.

"In most navies gun-metal is used for the hubs of propellers."

BOSS AND FLANGES ON BLADES.

Chase: "Diameter of flange = 2.25 × diameter of shaft.
Thickness of flange = .85 × diameter of stud for bronze or steel.
Length of boss = 2.7 × diameter of shaft.
Thickness of boss = .65 × thickness of blade at shaft axis.
Diameter of boss = 3.3 × diameter of shaft.
Taper of boss = 1″ to 1′.

"The fore and aft section of the boss should be oval—the principal radius being .8 × diameter of boss."

Seaton: "A well-designed and carefully made screw should have the base of the blade conforming to the general outline of the boss, and the nuts or bolt heads recessed into the blade base and covered with a metal case or cemented flush with the surface."

BOLTS.

Chase, p. 161: "The bolts are usually of naval brass (muntz-metal with the addition of a small percentage of tin), or are of one of the stronger bronzes for gun-metal or bronze propellers, and of mild steel for cast-iron or cast-steel propellers.

The size is from the formula: $a \times N \times r = \frac{T \times L}{K}$.

$a =$ area in sq. in. of one stud bolt at bottom of thread.
$N =$ No. of studs or bolts for one blade (usually 7 to 11).
$r =$ radius of circle of studs or bolts in ins.

$$T = \text{indicated thrust} = \frac{\text{I. H. P.} \times 33{,}000}{\text{mean pitch} \times \text{revolutions per min.}}.$$

$L = .6 \times$ total length of blade in feet (flange joint to tip).
$K = 17$ for steel bolts or studs.
14 for naval brass or bronze studs or bolts."

Another method, *Chase:* "To determine the size of bolts,

Let P be the I. H. P. per blade.
h be the distance of the center of pressure of the blade from the under side of the flange in feet.
K be a coefficient, varying from 18 to 21 for gun-metal.
R be the number of revolutions per minute.
B be the combined area in sq. ins. of the bolts at the bottom of their threads, on the face or pressure side of the blade.

Then, $B = \frac{P\,h\,K}{C\,R}$.

"The number of bolts having been determined, their diameter is found by dividing B by the number of bolts for the area of one bolt; the bolts on one side of the blade being generally of the same diameter for convenience."

The difficulty in the use of this formula is in determining accurately the distance of the center of pressure of the blade from the under side of the flange. It may be approximately determined and the formula used as a check.

TAPER OF SHAFT, FEATHER, THREAD, NUT.

Chase, p. 161: "The taper of the part of the shaft within the boss should be about one inch in diameter for each foot of length, but never less than three-fourths of an inch.

"The thread of the large nut that holds the propeller on the shaft is 2½ threads per inch, regardless of the diameter. It should be left handed when the propeller is right handed, and vice versa.

"The nut should be securely locked, preferably by a plate fixed to the after end of the boss by set screws.

"The propeller should be secured by one feather or key extending the whole length of the boss, the proportions of which may be:—

Breadth of key = .22 × largest dia. of shaft + .25″.
Thickness of key = .55 × breadth.

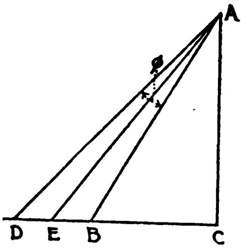
"The diameter of the screwed end of the shaft should be sufficiently reduced to allow the key to be fitted in from the after end clear of the thread."

The key should have clearance at the after end, so that the nut will not come tight against it, and should be partly imbedded in the shaft, the forward end being rounded.

In Bureau Drawings, sometimes the key is arranged to be secured in place in the shaft and the propeller slid on from aft, in which case the top of the thread on the shaft should be less in diameter than the smallest bore of the propeller.

ALTERING THE PITCH.

Barnaby, p. 91: "For the sake of possibe adjustments that may be desired on the trials of machinery, and also for finding the pitch when it becomes necessary to reduce the steam pressure in the boilers, it is customary to make the bolt holes in the blade flanges oval, in order that the inclination of the blades to the axis may be altered. The amount of the oval may be determined thus: "Let



the true pitch of the blade be 2π C E and let the desired range of pitch be 2π E B and 2π E D respectively on each side of this pitch. Take a radius C A, so that A is about half way up the blade. Join A D, A E, A B. Then, if $\phi =$ the angle D A B, the holes in the flange must be so elongated as to admit of the blade being turned through $\frac{\phi}{2}$ degrees on each side. The amount of this elongation on each side is, then, $\frac{\pi C}{360} . \frac{\phi}{2} = \pi C \frac{\phi}{720}$, ϕ being measured in degrees.

2 × this length + the diameter of the bolt or stud = length of slot. C = diameter of bolt circle."

CALCULATIONS REQUIRED.

Percentage of slip.
Pitch, mean.
Pitch, greatest.
Pitch, least.
Pitch, ratio.
Diameter from pitch ratio.
Depth of immersion (*Chase*).
Depth of immersion (*Practice*).
Distance below blades.
I. H. P. of auxiliaries.
I. H. P. of each engine used in calculations.
Value of K USED.
Diameter, *Seaton*.
Diameter, *Foley*.
Diameter, *Foley*, 2d method.
Diameter by comparison.
Average of methods.
Peripheral speed.
DIAMETER USED.
Value of K used.
Helicoidal area, *Seaton*.
Diameter of boss, *Foley*.

Diameter of boss, *Chase.*
Diameter of boss, *Chase,* 2d method.
DIAMETER OF BOSS, USED.
Disc area.
Helicoidal area from disc area.
Thrust, *Seaton.*
Thrust, *Foley.*
THRUST USED.
Helicoidal area from thrust, *Seaton.*
Helicoidal area from thrust, *Bureau.*
Helicoidal area, average.
HELICOIDAL AREA USED.
Blades, number.
Blades, material.
Bend back of blades.
Helicoidal area of each blade.
Major axis of trial ellipse.
Minor axis of trial ellipse.
Major axis of ellipse USED.
Minor axis of ellipse USED.
Length of blade, *Chase.*
Length of blade, *Seaton.*
LENGTH, USED.
Distance of max. breadth of blade from center.
Breadth of blade at the tip.
Thickness of blade at 1½ d, *Seaton,* 1st method.
Same, *Seaton,* 2d method.
Same, *Chase.*
THICKNESS OF BLADES, USED.
Thickness at tip, *Seaton.*
Same, *Chase,* 1st method.
Same, *Chase,* 2d method.
THICKNESS AT TIP, USED.
Boss, diameter.
Boss, length, *Seaton.*
Boss, length, *Chase.*
BOSS, LENGTH USED.

Boss, thickness, *Chase.*
Boss, taper.
Boss, radius.
Flange, diameter, *Chase.*
Flange, thickness, *Chase.*
Bolts, material, *Chase.*
Bolts, number, *Chase.*
Bolts, diameter, *Chase.*
Bolts, diameter, *Chase,* 2d method.
Taper of shaft.
Threads per inch on shaft.
Right or left handed.
Breadth of key.
Thickness of key.
Length of key.
Dia. of screwed end of shaft (outside).
Depth of thread.
No. of threads to 1″.
Diameter of plug in the end of shaft.
Length of same.
No. of threads to 1″ of same.
Length of slots in flanges of blade.

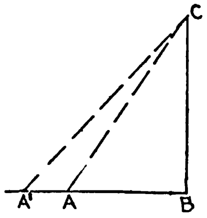
GEOMETRY OF THE SCREW.

It is first noted that the face only of the blade is drawn of the true geometrical form and is the driving face of the blade; the form of the back depends on the thickness given at the different radii.

An exception is sometimes made in the case of torpedo-boats, where rapid manœuvring is required and rapid speed astern, as in the case of the U. S. Torpedo-boat "Ericsson," where the front and back of the blades are alike.

Barnaby, p. 75: "If a point move on the surface of a cylinder in such a way that, while moving uniformly around the cylinder, it advances uniformly in the direction of its axis, it will trace a curve known as the helix. Imagine the cylinder cut on one side

by a straight line, B C, parallel to the axis meeting the helix in consecutive points A and C, and then unrolled and laid flat, the circumference through A will become a straight line A B; B C at right angles to it will represent the direction of the axis while that part of the helix formed during the complete revolution of the tracing point will be represented by the line A C. Since the distances moved by the point in the directions A B, B C are proportional, A C is a straight line. B C being the distance moved through in the direction of the axis, while the point goes once entirely round the cylinder, is the pitch, while the angle B A C, which the unrolled helix makes with the plane at right angles to the axis, is the angle of the helix or screw.



"If a straight line moves uniformly around an axis which it intersects, and to which it is always at right angles, advancing at the same time uniformly in the direction of the axis, it will sweep out a surface known as a helicoid, and every point in the generating line will describe a helix as shown above, necessarily lying on this helicoid. Since, during a complete revolution of the generating line, every point moves through the same distance in the direction of the axis, the helicoid is the surface of uniform pitch; that is, B C is constant for the helices of all points in the generating line. A helicoid can therefore be, and often is, used for the acting face of a screw blade of uniform pitch. It is not necessary, however, that the generating line should be at right angles to the axis; such a surface may be generated by any line, straight or curved, moving uniformly along and revolving uniformly around an axis, intersecting and always making the same angle with it.

"The helix traced by any point in the generating line will also be the curve of intersection with the screw surface of a coaxial cylinder of radius equal to the perpendicular distance of the point from the axis. The larger the radius of the cylinder, the larger of course the length of the circumference, as A′B; and, as the pitch is constant, it follows that the angle of the helix must decrease as the radius of the intersecting cylinder increases. We thus arrive at the fundamental geometrical property of a surface of uniform pitch, viz.: Co-axial cylinders intersect it in helices, all of which have the same pitch, but whose angles vary, decreasing as the radius of the cylinder increases. Near the axis, therefore, the helices will approximate in direction to that of the axis, and as the distance from the axis increases, they will lie more and more at right angles to it. If θ be the angle of the helix, p the pitch, and r the radius of the intersecting cylinder:—$\tan \theta = \frac{p}{2 \pi r}$."

Barnaby, p. 78: "The expanded blade area, which may be described as a flat surface of approximately equivalent area to that of the blades, both as to amount and disposition, is derived as follows:

"A co-axial cylinder will intersect the screw surface in a helical curve making a certain angle with the axis, and it will intersect a plane passing through that diameter of the cylinder which passes through the middle point of the helical curve and making the same angle with the axis, in an elliptical arc. The length of the screw being small compared with the pitch, these two arcs will nearly coincide, and no great error will be involved by assuming that they do coincide. Imagine these elliptical arcs at all radii to be swung around a common center line till they all lie in the same plane with their major and minor axes respectively coincident (though necessarily of different lengths), then the curve passing through their extremities will form the expanded blade area. This area is very nearly equal to the actual working face of the blade, being in fact somewhat less than it."

DRAWING.

The following is a general form for laying down the screw to be observed in the designing-room of the Department of Steam Engineering at the United States Naval Academy:—

The drawing is to consist of vertical elevation of the propeller from aft; one-third of the hub and one flange of the blade in

section; a side elevation; a side elevation of one blade showing the section of the hub and the thickness of the blade; a drawing of the developed area and sections of the blade either on the

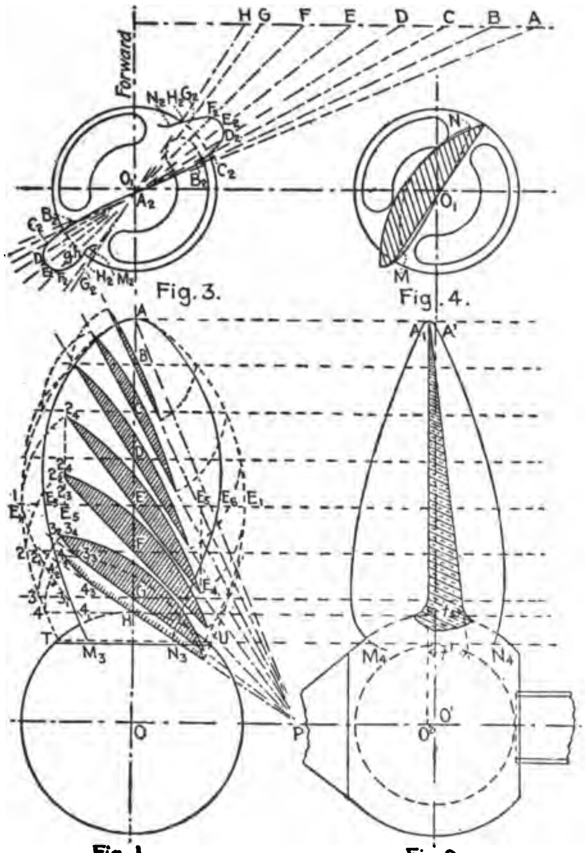


Fig. 3. Fig. 4.

Fig. 1. Fig. 2.

Sheet 1.

angular lines or at right angles to the central line of the blade on one of the other figures; guide-irons and sections of flanges and bolts as required.

The side elevation of the blade showing the sections of the hub may be combined with the side elevation of the whole propeller by making part section and part elevation, though it is better to separate them for clearness.

Having determined most of the dimensions from the previous notes, we first begin Fig. 1, Sheet 1, a vertical elevation of the blade looking forwards. Draw the center line O P O′ representing a line at right angles to the center line of the shaft, O. Lay off the circle showing the greatest diameter of the hub, and draw a vertical center line through O, the center of the shaft, for the center line of the blade. Lay off O A equal to the radius of the propeller. Lay off equally on each side of this line the developed area of one blade required. Since about one-fifth of this area will be inclosed in the hub, it is well to lay off at first the area on O A as an axis equal to six-fifths of the area required. This area may be laid off entirely by trial suiting the curves to the ideas of the designer, but it is better to use as a basis for a trial area an ellipse with O A as a major axis and a minor axis found as follows:—

Area of an ellipse $= \pi a b$. In this case, the area must equal $\frac{6}{5} \times$ developed area calculated for one blade. The major axis is equal to O A. To find, then, the minor axis, we have $\pi a b = \frac{6}{5} \times$ helicoidal area of one blade.

Then, $$\frac{6 \times \text{helicoidal area of one blade}}{5 \pi a} = b.$$

The ellipse is then constructed according to any approximate method, and the area $T E_1 A E_1 U H T$ is measured by planimeter. If the area does not agree with that required, it may be enlarged or diminished by changing the length of the minor axis until the required area is obtained. If the area be too small, one way of adding to the surface is to enlarge the lower part, thus making the blade more pear-shaped, according to the ideas of the designer. If the area be too large, sometimes a portion is removed from the top of the blade, thus again producing a pear-shaped blade.

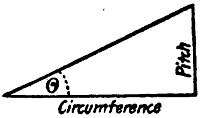
In Fig. 1, the lower part of the blade has been made pear-

shaped, the lower portion of the original ellipse not being drawn.

Some designers use for the semi-major axis of the ellipse about 85 per cent. of the radius of the propeller, and take four-tenths or more, as required, for the semi-minor axis. The ellipse then is not so much within the circle of the hub and a less proportion is cut off.

Divide the center line H A into any number of parts—five or six, and preferably equal, or, as generally constructed, make the outer divisions, A B, B C, etc., to F G equal and a definite number of feet and half feet long and the divisions G H nearest to the hub longer or shorter than the others, if necessary. Draw horizontal lines through these points.

Lay off $O P = \frac{\text{pitch}}{2\pi}$ and draw lines through P A, P B, etc., to P H. $\operatorname{Tan} \theta = \frac{\text{pitch}}{\text{circumference}} = \frac{\text{pitch}}{2 \pi r} = \frac{O P}{O A}$.



r is a varying quantity, O E, O D, etc., to O A. Since the tangent of the angle which any one of these lines P A, P B, etc., passing through a point in O A distant r from O makes with A O is $\frac{p}{2 \pi r}$, these lines represent the angles of the blade of the various sections at distances O B, O C, etc., from the axis.

From the points B, C, D, etc., set off on the lines P B, P C, etc., the lengths of the sections made by the horizontal lines through B, C, D, etc., on the expanded blades, as shown by the arcs.

Next, in Fig. 2, Sheet 1, the side elevation, or fore and aft projection, draw the vertical center line O′ A′ for the center line of the hub. The blade is seldom laid off on this line, as it is frequently found that, if the thickness of the blade be laid off from this line, the back of the blade will interfere with the bolts used for securing the blade to the hub. Consequently, the working face is moved back, as the thickness is placed on the forward side. This distance of moving back the center line of the blade varies with the diameter of the propeller, and in this case is 1½″. The line $O^2 A_1$, 1½″ back, is then the center line of the blade.

From previous calculations we have the thickness of the blade at the hub and at the tip and at a distance of 1½ × diameter of the shaft from the center of the shaft. Lay off from $O^2 A_1$ these distances (the one at the hub, t, and the one at 1½ dia. of shaft, t′, will be checks on each other), and connect the points of thickness at tip and hub by a straight line, the extension of this line (dotted) to the thickness at 1½ d, being drawn.

Where the line showing the center line of the blade and the one showing the thickness meet the hub easy curves are drawn as shown.

We have then a section of the blade through the center line of the developed blade.

This thin sectional strip showing the thickness at the different distances from the center is supposed to represent a section through the center line of the flattened or developed blade, as it is apparent that only on this assumption could a plane section be obtained giving the thickness of metal normal to the blade.

Extend the horizontal lines through A, B, etc., of Fig. 1, Sheet 1, to Fig. 2, which gives the thickness of the blade for the various sections represented by the horizontal lines. Transfer these thicknesses to Fig. 1 by laying them off at B, C, etc., perpendicular to the lines representing the angles of the blades, as shown at E and F, and draw through these points arcs of circles touching the edges of the sections already obtained. These arcs should really be portions of ellipses, but are so near portions of circles that they are practically correct. This will give the shapes of the sections of the blade at different radii.

In some drawings these sections are laid out horizontally on the lines B, C, etc. Either way may be followed.

Those sections near the base are sometimes modified in order to allow the water to escape freely from the following edge of the blade, in which case the outlines of these sections are not finished until the amount of this modification is determined. The following edge is frequently curved upwards or back, and the amount

of this flare is governed by the designer. The portion to be rounded back is shown by the assumed dotted line $1 2_1 3_1$ on the developed blade. By measuring the distances $2 2_1$, $3 3_1$, $4 4_1$, and setting them off as $2_2\,2_3$, $3_2\,3_3$, $4_2\,4_3$, on the lines showing the angles at the different distances from the center, we find the points where the blades begin to flare. Draw easy curves $2_3\,2_4$, $3_3\,3_4$, $4_3\,4_4$, thus dropping back the working edge near the hub and still preserving about the same helicoidal area. The perpendicular distances from 2_4, 3_4 and 4_4 to the corresponding lines for the angles of the blades give the actual amounts of the setting back of the edges to be used in the plan view.

These points 2_4, 3_4 and 4_4 are connected by arcs of circles with the points showing the thickness of the blade at the center and the other edge as found.

The plan, Fig. 3, Sheet 1, is next begun. A circle of the diameter calculated for the flange of the blade is drawn. If the working face of the blade were not moved back from the center line of the hub, Fig. 2, the projection of the central line of the blade would be at O_1, Fig. 3. In this case, the center having been moved back, the center of the blade in the plan view will be at A_2, a distance aft equal to that in Fig. 2. Through the point A_2 draw lines representing the angles of the blade for the sections A, B, C, etc., as in Fig. 1, and lay off on these lines the developed lengths at the different sections. For instance, $E_1\,E\,E_1 = E_2\,A_2\,E_2$. Note that the lengths of the sections of the lower portions of the blade where cut away are measured to the actual length on the after edge of the blade, as $A_2\,G_2 = G\,3_2$. Then lay off perpendicular to $A_2\,G_2$ the distance from the section, Fig. 1, that the blade is bent back from the section, as $G_2\,g = 3_2\,3_4$.

Before completing the plan view, the intersection of the blade with the spherical surface of the flange is obtained. In Fig. 1 it is seen that the lower edges of the developed blade, T and U are just within the circumference of the flange, so that the edges of the intersection of the flange and hub should be on this line. To get the intersection it is convenient to draw Fig. 4, though the same operation may be carried on in Fig. 3. Draw the circle of the flange, Fig. 4, and place in its proper position the section of the blade at H of Fig. 1. This section is, of course, tangent to the round up of the flange. It is seen that this section overlaps the circle of the flange, so that from this point downwards to the blade the width must be gradually shortened, so that the edges may fall within the circumference of the flange; and these edges must be distant from the center of the flange a distance equal to the radius at T U, Fig. 1. Another consideration that affects the positions of the points of intersection of the edges of the flange and the blade is the position of the bolts, covering plates, etc. These are drawn in, according to experience. The number of bolts has already been decided upon and also their diameters, so that the distance apart and the clearance must be determined to the best advantage. The outlines of the covering plates being determined, the section at H is worked down in Fig. 4, so as to shorten and possibly twist a little one way or the other as required. Drawings from the Bureau of Steam Engineering are to be consulted for this. The points M and N will be determined from the required distance from the center, and the best arrangement of the line of intersection of the flange and blade as shown by the unhatched lines, taking care that the greatest thickness at the middle point of the intersection is not less than the corresponding thickness of the section H.

This theoretical intersection is not shown on the final drawing, as the fillets at the bottom of the blade cause easy curves of intersection.

To find where the ends of the blades intersect the flange in the different projections:

The points M and N are transferred to Fig. 3, thus giving M_2 and N_2, and are projected to Fig. 1 to the line T U, thus giving M_3 and N_3. Project M and N, Fig. 4, to a vertical line and measure the distances from the center of the blade and transfer these distances on either side of $A_1 C_2$, Fig. 2, on the line T U extended for the points M_4 and N_4.

To obtain the projections of the working edge of the blade:

Having determined M_2 and N_2, Fig. 3, we have now all the points of the plan, so we may draw in the curve $N_2 E_2 A_2 E_2 M_2$.

In Fig. 1, project from Fig. 3 the ends of the lines E_2 A_2 E_2, etc., to their corresponding lines on Fig. 1 and draw a curve through these points and extend to M_3 and N_3, taking care to note which edge falls behind the round up of the flange.

For Fig. 2 project the ends of the angular lines of Fig. 3 to a vertical plane and measure on Fig. 2 from the center line A_1 O^2 and on the corresponding lines the distances on this vertical plane from the projection of the center A_2. Connect the points so found with a curve, extending it to M_4 and N_4, noting which curve falls behind the round up of the flange.

Another method of finding the projections is as follows:

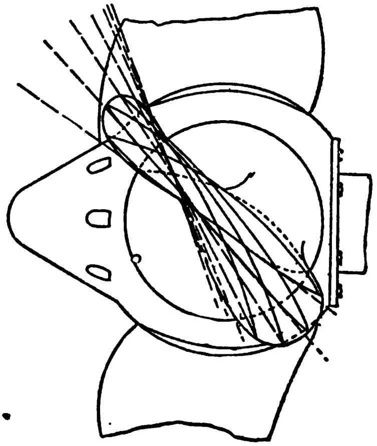
In Fig. 1 project E_4 vertically to E_5. Measure E_5 E_4 from E to E_6. This will be then one of the points of the intersection. In Fig. 1 measure E E_5 and lay it off on Fig. 2 on the corresponding line and edge of the blade from the center line O^2 A_1 of the blade.

The same methods are followed for all points of the projections.

OUTLINE OF THE BACK OF THE BLADE.

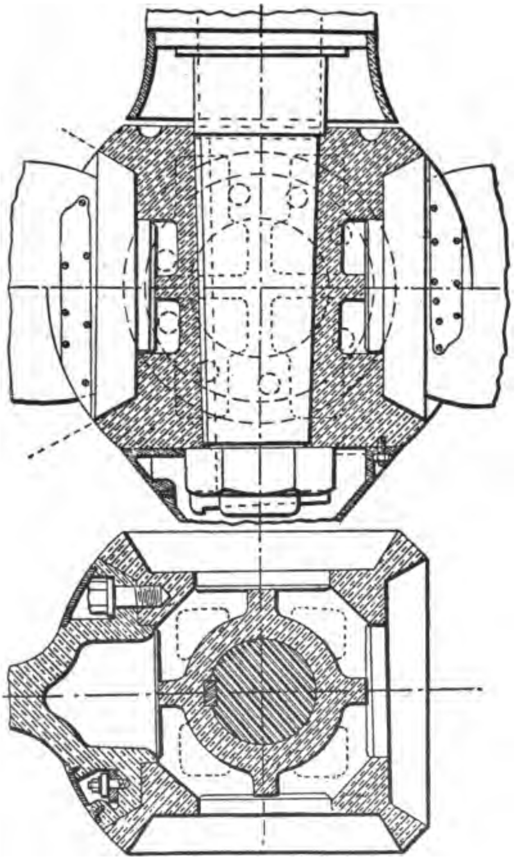
In the plan, Fig. 3, Sheet 1, it is evident that a portion of the back of the blade is seen. The outline is obtained as follows: On the elements B_2 B_2, C_2 C_2, etc., lay off the thickness strips as determined in Fig. 1. Then the highest portions of the different thicknesses will be points in the outline of the view of the back of the blade, so that a curve drawn tangent to these curves will give the outline of the back of the blade.

Sheet 2 shows this method of finding the outline of the back of the blade. At some point this outline gradually joins the curve of the flange. This is represented approximately by a curve.

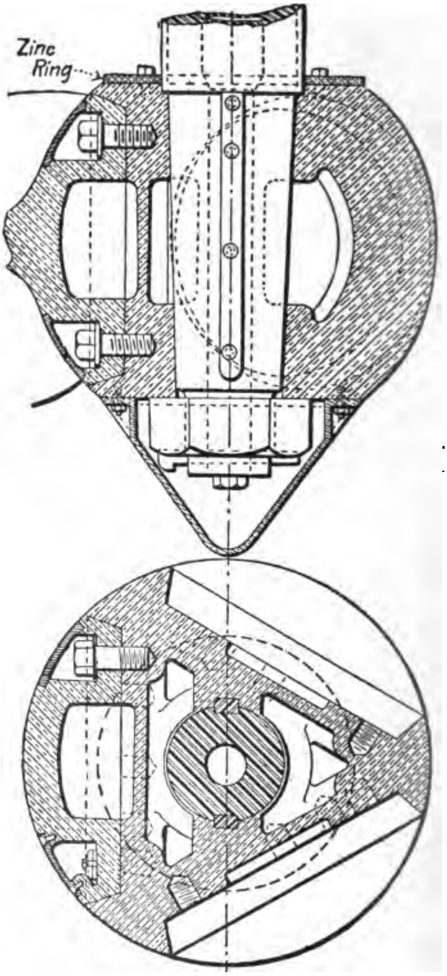


SHEET 2.

Having determined the general outline of the blades for the different views, the hub may be drawn and finished according to the dimensions and details as determined upon. Some of the methods as used in the designs of machinery for vessels of the United States Navy are shown in Sheets 3 and 4.



Sheet 3.

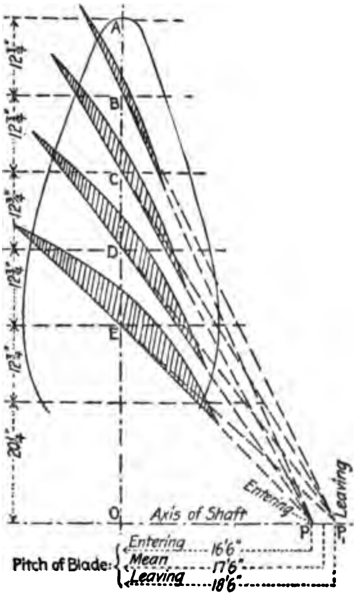


Sheet 4.

In the case of a screw with expanding pitch the above method of drawing the propeller is somewhat changed. See Sheet 5.

Divide the center line as before and set off $O\,P = \dfrac{\text{entering pitch}}{2\,\pi}$ and $O\,P' = \dfrac{\text{leaving pitch}}{2\,\pi}$. From P draw lines through B, C, etc., these lines giving the angles of the blades for the entering edges. From P′ draw lines through B, C, etc., for the angles of the blade for the leaving edge. These lines intersect at the center, and for the entering half of the blade the lines from P form the bases of the sections cut out, while lines from P′ form the bases of the sections cut from the after half of the blade as shown.

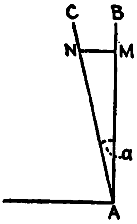
The rest of the drawing conforms to this method.



SHEET 5.

GENERATRIX INCLINED.

Barnaby, p. 85: "Blades are sometimes made with the generating line inclined to the axis; or, in technical terms, they are made with a skew. Let the two straight lines, A B, A C, of the figure, the former at right angles to the axis, the latter inclined to A B at an angle *a,* moving together, generate screw surfaces of uniform and equal pitch. Then the helices of intersection of these two surfaces will be exactly similar, and one will be always a constant distance from the other; this distance being at a radius r, equal to r tan *a.* Imagine these two surfaces so far similar that when A C at any radius leaves the surface, A B at the same radius leaves its surface, then the expanded area of the sur-



face so formed by A B will represent what may be termed the effective expanded area of that formed by A C, and this should be of the elliptical or other form which would have been used if the generating line had been at right angles to the axis. A blade generated by A C would therefore be formed from a blade generated by A B, simply by setting the helices of intersection definite distances aft; the distance at M, for example, being M N. It follows, therefore, that the athwartship projection of the blade

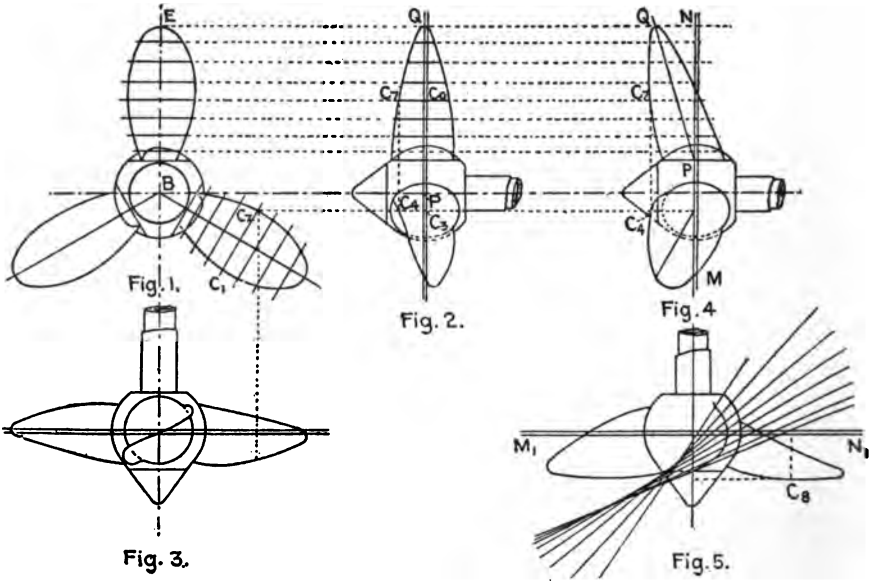


Fig. 1. Fig. 2. Fig. 4 Fig. 3. Fig. 5.

Sheet 6.

for the same "effective" expanded area is independent of the skew, and consequently for a skew blade with effective expanded area as for the blades shown in Figs. 1, 2 and 3, Sheet 6, the athwartship projection will be as in Fig. 1, no matter what the skew may be:

"The fore and aft projection of the upper blade, Fig. 4, will be formed by using a center line inclined to the vertical at an angle equal to the inclination to the vertical of the generating line, and proceeding as in Fig. 2, Sheet 1, setting off the distances

horizontally. The projections of the lower blades are determined in a similar way to that described before. For example, the point whose athwartship projection is c_2, Fig. 1, Sheet 6, will appear on Fig. 4 as c_4, lying in the same vertical line as c_7 (c_7 being the position of this point when the blade is upright), and is perpendicularly away from the axis, a distance equal to that of c_2 from the horizontal plane through the axis.

"For the horizontal projections, Fig. 5.—For the top blade the pitch lines are not all drawn through the same point as in Fig. 3, but each line is drawn at a corresponding angle to the axis through a point on the axis at a distance from $M_1 N_1$, equal to the distance of the corresponding point on the generating line P Q from M N, Fig. 4, the process then being as previously shown. For the lower blade we proceed as follows:—The point (c_2, Fig. 1) corresponding to c_7, Fig. 4, when the blade is upright, will appear in Fig. 5 as c_8 at a distance from the axis equal to the distance of c_2, Fig. 1, from the vertical plane through the axis and from $M_1 N_1$ equal to the distance of c_7 from M N, Fig. 4.

"The projections of a three-bladed propeller with skew blades are shown in Figs. 1, 4 and 5, except that the left-hand lower blade of Fig. 1 has not been shown in Fig. 4 to avoid confusion.

"Where the generating line is curved, the method is now obvious."

USEFUL TABLES.

Standard Dimensions of Bolts and Nuts for the United States Navy.

Diameter.		Area		Thr'ds.	Long Diameter.		Short D.	Depth.	
Nom.	Effect.	Nom.	Effect.	No.	Hex.	Square.	H. & Sq.	Head.	Nut.
1/4	.185	.049	.026	20	9/16	23/32	1/2	1/4	1/4
5/16	.240	.077	.045	18	11/16	27/32	19/32	19/64	5/16
3/8	.294	.110	.067	16	25/32	31/22	11/16	11/32	3/8
7/16	.345	.150	.093	14	29/32	1 3/32	25/32	25/64	7/16
1/2	.400	.196	.125	13	1	1 1/4	7/8	7/16	1/2
9/16	.454	.249	.162	12	1 1/8	1 3/8	31/32	31/64	9/16
5/8	.507	.307	.202	11	1 7/32	1 1/2	1 1/16	17/32	5/8
3/4	.620	.442	.302	10	1 7/16	1 3/4	1 1/4	5/8	3/4
7/8	.731	.601	.419	9	1 21/32	2 1/32	1 7/16	23/32	7/8
1	.837	.785	.550	8	1 7/8	2 5/16	1 5/8	13/16	1
1 1/8	.940	.994	.694	7	2 3/32	2 9/16	1 13/16	29/32	1 1/8
1 1/4	1.065	1.227	.891	7	2 5/16	2 27/32	2	1	1 1/4
1 3/8	1.160	1.485	1.057	6	2 17/32	3 3/32	2 3/16	1 3/32	1 3/8
1 1/2	1.284	1.767	1.294	6	2 3/4	3 11/32	2 3/8	1 3/16	1 1/2
1 5/8	1.389	2.074	1.515	5 1/2	2 31/32	3 5/8	2 9/16	1 9/32	1 5/8
1 3/4	1.491	2.405	1.746	5	3 3/16	3 7/8	2 3/4	1 3/8	1 3/4
1 7/8	1.616	2.761	2.051	5	3 13/32	4 5/32	2 15/16	1 15/32	1 7/8
2	1.712	3.142	2.302	4 1/2	3 19/32	4 13/32	3 1/8	1 9/16	2
2 1/4	1.962	3.976	3.023	4 1/2	4 1/32	4 15/16	3 1/2	1 3/4	2 1/4
2 1/2	2.176	4.909	3.719	4	4 15/32	5 15/32	3 7/8	1 15/16	2 1/2
2 3/4	2.426	5.940	4.622	4	4 29/32	6	4 1/4	2 1/8	2 3/4
3	2.676	7.069	5.624	4	5 11/32	6 17/32	4 5/8	2 5/16	3
3 1/4	2.926	8.296	6.724	4	5 25/32	7 1/16	5	2 1/2	3 1/4
3 1/2	3.176	9.621	7.922	4	6 7/32	7 19/32	5 3/8	2 11/16	3 1/2
3 3/4	3.426	11.04	9.219	4	6 5/8	8 1/8	5 3/4	2 7/8	3 3/4
4	3.676	12.57	10.61	4	7 1/16	8 21/32	6 1/8	3 1/16	4
4 1/4	3.926	14.19	12.11	4	7 1/2	9 3/16	6 1/2	3 1/4	4 1/4
4 1/2	4.176	15.90	13.70	4	7 15/16	9 23/32	6 7/8	3 7/16	4 1/2
4 3/4	4.426	17.72	15.39	4	8 3/8	10 1/4	7 1/4	3 5/8	4 3/4
5	4.676	19.64	17.17	4	8 13/16	10 25/32	7 5/8	3 13/16	5
5 1/4	4.926	21.65	19.06	4	9 1/4	11 5/16	8	4	5 1/4
5 1/2	5.176	23.76	21.04	4	9 11/16	11 27/32	8 3/8	4 3/16	5 1/2
5 3/4	5.426	25.97	23.12	4	10 3/32	12 3/8	8 3/4	4 3/8	5 3/4
6	5.676	28.27	25.30	4	10 17/32	12 29/32	9 1/8	4 9/16	6

DECIMALS AND FRACTIONS OF AN INCH.

1/32	.031	9/32	.281	17/32	.531	25/32	.781
1/16	.062	5/16	.312	9/16	.562	13/16	.812
3/32	.094	11/32	.344	19/32	.594	27/32	.844
1/8	.125	3/8	.375	5/8	.625	7/8	.875
5/32	.156	13/32	.406	21/32	.656	29/32	.906
3/16	.188	7/16	.438	11/16	.688	15/16	.938
7/32	.219	15/32	.469	23/32	.719	31/32	.969
1/4	.250	1/2	.500	3/4	.750	1	1.000

TABLE OF AREAS OF CIRCLES, ADVANCING BY EIGHTHS.

Diam.		+⅛″	+¼″	+⅜″	+½″	+⅝″	+¾″	+⅞″
0″	.000	.012	.049	.110	.196	.307	.441	.601
1″	.785	.994	1.227	1.485	1.767	2.074	2.405	2.761
2″	3.142	3.547	3.976	4.430	4.909	5.412	5.940	6.492
3″	7.069	7.670	8.296	8.946	9.621	10.32	11.05	11.79
4″	12.57	13.36	14.19	15.03	15.90	16.80	17.73	18.67
5″	19.64	20.63	21.65	22.69	23.76	24.85	25.97	27.11
6″	28.27	29.46	30.68	31.92	33.18	34.47	35.73	37.12
7″	38.48	39.87	41.28	42.72	44.18	45.66	47.17	48.71
8″	50.27	51.85	53.46	55.09	56.74	58.43	60.13	61.86
9″	63.62	65.40	67.20	69.03	70.88	72.76	74.66	76.59
10″	78.54	80.52	82.52	84.54	86.59	88.66	90.76	92.89

VALUES OF $\frac{d^2}{16}$, WHEN d ADVANCES BY EIGHTHS.

d		+⅛″	+¼″	+⅜″	+½″	+⅝″	+¾″	+⅞″
0″	.0000	.0010	.0039	.0088	.0156	.0244	.0352	.0479
1″	.0625	.0791	.0977	.1182	.1406	.1650	.1914	.2197
2″	.2500	.2822	.3164	.3526	.3906	.4307	.4727	.5166
3″	.5625	.6104	.6596	.7119	.7656	.8213	.8789	.9385
4″	1.000	1.064	1.129	1.196	1.266	1.337	1.410	1.485
5″	1.563	1.642	1.723	1.806	1.891	1.978	2.066	2.157
6″	2.250	2.345	2.441	2.540	2.641	2.743	2.848	2.954
7″	3.063	3.173	3.285	3.399	3.516	3.634	3.754	3.876
8″	4.000	4.136	4.254	4.384	4.516	4.649	4.785	4.923
9″	5.063	5.204	5.348	5.493	5.641	5.777	5.941	6.095
10″	6.250	6.407	6.566	6.727	6.891	7.056	7.222	7.391

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